TRANSACTIONS

VOLUME 47

MILWAUKEE MEETING
NEW YORK MEETING
1925

PUBLISHED BY
THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
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1926

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GIFT
DEAN M. E. COOLEY

PUBLICATIONS COMMITTEE'S FOREWORD

THIS volume of TRANSACTIONS contains chiefly papers which formed part of the Spring and Annual Meeting programs, with a few others presented at regional and sectional gatherings. Material which the Committee believes to be of permanent reference value has been selected, and both papers and discussions have been subject to thorough editing and revision since their presentation.

Prompt selection of Spring Meeting papers to be included in TRANSACTIONS made it possible to proceed with editorial work immediately following the Milwaukee meeting, and to complete presswork upon that part of the volume before the end of the year. This not only cleared the way for work upon the Annual Meeting papers as soon as the meeting was over, but expedited the issuance of pamphlet copies of the Spring Meeting papers and discussions.

Attention is called to a procedure inaugurated in the 1924 volume and followed in this one. Regional, Spring, and Annual Meeting papers printed in *Mechanical Engineering* but not in this volume, technical reports, and other important contributions to the *Journal* during 1925, are listed on pages immediately following the main index to this volume.

At the request of the Council a letter ballot was taken early in the year to ascertain whether the majority of the members of the Society desired to receive TRANSACTIONS. Of 10,010 members who replied, 8,216 stated that they did not wish to forego this privilege, and the Council therefore took action to continue the issuance of TRANSACTIONS in its usual form.

O. G. DALE, *Chairman*
J. T. WILKIN
R. E. FLANDERS
K. H. CONDIT
E. D. DREYFUS
Publications Committee

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[1] The Society shall not be responsible for statements or opinions advanced in papers or in discussion at meetings of the Society or of its Divisions or Sections, or printed in its publications.

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WILLIAM F. DURAND

PRESIDENT OF THE SOCIETY FOR 1924-1925

WILLIAM F. DURAND was born in Bethany, Connecticut, on March 5, 1859, and was graduated in 1880 from the United States Naval Academy. For the next seven years he was in Naval Service, serving in the Engineers Corps. In 1887 he resigned to take up teaching as a profession, first at the Michigan State Agricultural College where he was professor of mechanical engineering until 1891, and then at Cornell where he was professor of marine engineering until 1904. In 1904 he went to Stanford University where he was professor of mechanical engineering until June, 1924, when he retired from active service and was placed on the emeritus roll.

Dr. Durand's most noteworthy service, possibly, was rendered to the country at the time of the war. In 1917 he was in Washington as chairman of the National Advisory Committee for Aeronautics (of which Committee he has been a member since 1915), and vice-chairman of the National Research Council Committee on Engineering. The following year he spent in Paris as Scientific Attaché to the United States Embassy, representing the National Research Council, and charged with the duty of effecting and maintaining an interchange between Paris and Washington of technical and scientific information regarding war problems, and also as a member of the Interallied Commission on War Inventions.

In 1925 he was appointed by President Coolidge a member of the President's Aircraft Board and served that Board as member and Secretary during the three months of its work in Washington.

As an inventor Dr. Durand is best known for his three-point caliper, for determining true circular contours, and his radial planimeter for averaging ordinates in polar diagrams. As an author he is known for three books, Resistance and Propulsion of Ships (1898), Practical Marine Engineering (1901), and Hydraulics of Pipe Lines (1921), in addition to possibly a hundred technical papers published in governmental reports and the technical press.

Dr. Durand has been a member of the Society since 1883 and was a Vice-President from 1911 to 1913 and President for the year 1924-1925. He has long been active in the San Francisco Local Section and was its chairman in 1919-1920. He is also a member of the American Institute of Electrical Engineers, the American

Society of Naval Architects and Marine Engineers, the American Society of Naval Engineers (gold medal and life membership in 1899), the Society Technique Maritime, the National Academy of Sciences, the American Philosophical Society, the American Academy of Sciences, and the Royal Aeronautical Society.

No. 1952

SOCIETY AFFAIRS

ORGANIZATION AND MEMBERSHIP

ON THE following pages are given the names of those who made up the executive and administrative personnel of the Society, its representatives on joint activities, and a summary of its membership for the year 1925. The personnel of professional committees and divisions, Local Sections officers, and detailed information concerning the organization of the Society was printed in the Year Book for 1925.

OFFICERS AND COUNCIL

PRESIDENT

W. F. DURAND.....New York, N. Y.

VICE-PRESIDENTS

Terms Expire December, 1925

GEORGE I. ROCKWOOD.....Worcester, Mass.
W. J. SANDO.....Milwaukee, Wis.
H. BIRCHARD TAYLOR.....Philadelphia, Pa.

Terms Expire December, 1926

ROBERT W. ANGUS.....Toronto, Canada
SHERWOOD F. JETER.....Hartford, Conn.
THOS. L. WILKINSON.....Davenport, Iowa

MANAGERS

Terms Expire December, 1925

A. G. CHRISTIE.....Baltimore, Md.
JAMES H. HERRON.....Cleveland, Ohio
ROY V. WRIGHT.....New York, N. Y.

Terms Expire December, 1926

E. O. EASTWOOD.....Seattle, Wash.
E. R. FISH.....St. Louis, Mo.
FRANK A. SCOTT.....Cleveland, Ohio

Terms Expire December, 1927

JOHN H. LAWRENCE.....New York, N. Y.
EDWARD A. MULLER.....Cincinnati, Ohio
PAUL WRIGHT.....Birmingham, Ala.

PAST-PRESIDENTS

FRED J. MILLER (1925) ..... Harrisburg, Pa.
E. S. CARMAN (1926) ..... Cleveland, Ohio
DEXTER S. KIMBALL (1927) ..... Ithaca, N. Y.
JOHN LYLE HARRINGTON (1928) ..... Kansas City, Mo.
FRED R. LOW (1929) ..... New York, N. Y.

TREASURER

WM. H. WILEY[1] ..... New York, N. Y.

SECRETARY

CALVIN W. RICE ..... New York, N. Y.

STANDING ADMINISTRATIVE COMMITTEES

FINANCE

H. V. COES, *Chairman and Representative on Council* (1929)
ERIK OBERG, *Vice-Chairman* (1928) ARTHUR L. RICE (1926)
W. S. FINLAY, JR. (1925) HUGO DIEMER (1927)
Council Representatives: GEORGE I. ROCKWOOD (1925)
ROY V. WRIGHT (1926)

CONSTITUTION AND BY-LAWS

ROBERT I. CLEGG, *Chairman and Representative on Council* (1925)
T. E. COON (1926) E. E. HOWARD (1928)
CHARLES H. BROMLEY (1927) W. D. ENNIS (1929)

MEMBERSHIP

F. A. WALDRON, *Chairman and Representative on Council* (1925)
C. R. PLACE (1926) HOSEA WEBSTER (1928)
C. E. GORTON (1927) S. D. COLLETT (1929)

MEETINGS AND PROGRAM

L. B. MCMILLAN, *Chairman and Representative on Council* (1925)
C. N. LAUER (1926) R. M. GATES (1928)
E. HOWARD REED (1927) S. W. DUDLEY (1929)

PUBLICATIONS

O. G. DALE, *Chairman and Representative on Council* (1926)
J. T. WILKIN (1925) K. H. CONDIT (1928)
R. E. FLANDERS (1927) E. D. DREYFUS (1929)

LOCAL SECTIONS

CHARLES PENROSE, *Chairman and Representative on Council* (1925)
WILLIAM A. HANLEY (1926) JAMES D. CUNNINGHAM (1928)
JAMES A. HALL (1927) PAUL DOTY (1929)

AWARDS AND PRIZES

IRA N. HOLLIS, *Chairman and Representative on Council* (1925)
R. S. RILEY (1926) L. P. ALFORD (1928)
R. H. FERNALD (1927) A. M. GREENE, JR. (1929)

[1] Deceased May 2, 1925.

NOTE: Dates in parentheses denote expiration of term.

PROFESSIONAL DIVISIONS

F. O. Hoagland, *Chairman and Representative on Council* (1925)
H. V. Coes (1926) R. T. Kent (1928)
John H. Lawrence (1927) Archibald Black (1929)
Selby Haar, *Secretary*

LIBRARY

H. A. Lardner, *Chairman and Representative on Council* (1927)
W. W. Macon (1925) Percy H. Thomas (1928)
W. C. Wetherill (1926) The Secretary

PROFESSIONAL CONDUCT

Fred J. Miller, *Chairman and Representative on Council* (1925)
Chas. L. Newcomb (1926) Chas. R. Main (1928)
Edw. N. Trump (1927) I. E. Moultrop (1929)

EDUCATION AND TRAINING FOR THE INDUSTRIES

John T. Faig, *Chairman and Representative on Council* (1927)
Ira N. Hollis (1925) R. L. Sackett (1928)
D. C. Jackson (1926) S. S. Edmands (1929)

RELATIONS WITH COLLEGES

D. Robert Yarnall, *Chairman*
Wm. H. Kenerson W. H. Kavanaugh
L. C. Marburg H. G. Tyler

SPECIAL ADMINISTRATIVE COMMITTEES

ORGANIZATION OF THE SOCIETY

W. S. Finlay, Jr., *Chairman* James Partington
Dexter S. Kimball H. V. Coes

NOMINATING COMMITTEE

The members of the Nominating Committee representing the seven groups into which the membership of the Society has been divided, were elected at the 1924 Annual Meeting:

GROUP I.....C. K. Decherd, Meriden
W. R. Webster, Bridgeport, *Alternate*
GROUP II.....Jos. J. Nelis, New York
V. M. Frost, Newark, *Alternate*
GROUP III.....O. P. Hood, *Chairman*, Washington, D. C.
J. G. Hatman, Philadelphia, *Alternate*
GROUP IV.....R. R. Jones, Akron
E. G. Bailey, Cleveland, *Alternate*
GROUP V.....Thos. N. Wynne, Indianapolis
Arthur L. Rice, Chicago, *Alternate*
GROUP VI.....P. F. Walker, *Secretary*, Lawrence, Kan.
Wm. G. Christy, St. Louis, *Alternate*
GROUP VII.....C. I. Carpenter, Spokane
U. B. Hough, Spokane, *Alternate*

Note: Dates in parentheses denote expiration of term.

LOCAL SECTIONS IN NOMINATING COMMITTEE GROUPS

GROUP I

Boston
Bridgeport
Green Mountain (Vt.)
Hartford
Meriden
New Britain
New Haven
Providence
Waterbury
Western Mass.
Worcester

GROUP II

Metropolitan (N. Y.)
Foreign Members

GROUP III

Atlanta
Baltimore
Birmingham
Carolinas
Central Pa.
Chattanooga
Knoxville
Lehigh Valley
Louisville
Memphis
Philadelphia
Plainfield
Savannah
Virginia
Washington, D. C.

GROUP IV

Akron
Buffalo
Cleveland
Columbus
Eastern New York
Erie
Ontario
Pittsburgh
Rochester
Syracuse
Toledo
Utica

GROUP V

Chicago
Cincinnati
Detroit
Indianapolis
Milwaukee
Minneapolis
St. Paul
Tri-Cities

GROUP VI

Colorado
Houston
Kansas City
Mid-Continent
Nebraska
New Orleans
North Texas
St. Louis

GROUP VII

Inland Empire
Los Angeles
Oregon
San Francisco
Utah
Western Washington

TELLERS OF ELECTION OF OFFICERS

David B. Porter, *Chairman*
Graham L. Montgomery
Charles A. Howard

BIOGRAPHY ADVISORY

W. F. M. Goss, *Chairman*
Fred R. Low
R. V. Wright
Geo. A. Orrok
John R. Freeman

INDUSTRIAL AWARDS

Fred R. Low
F. A. Waldron
Ira N. Hollis
L. B. McMillan

STANDING PROFESSIONAL COMMITTEES

RESEARCH

R. J. S. Pigott, *Chairman and Representative on Council* (1928)
F. G. Hechler (1925)
Albert Kingsbury (1926)
D. R. Yarnall (1927)
A. E. White (1929)

STANDARDIZATION

E. C. Peck, *Chairman and Representative on Council* (1925)
C. M. Chapman (1926)
C. F. Hirshfeld (1927)
A. M. Houser (1928)
C. P. Bliss (1929)

POWER TEST CODES

Fred R. Low, *Chairman*

Terms Expire November 30, 1925
Fred R. Low
L. P. Breckenridge
R. H. Fernald
C. F. Hirshfeld
R. J. S. Pigott

Terms Expire November 30, 1926
A. G. Christie
Paul Diserens
C. E. Lucke
E. F. Miller
Geo. A. Orrok

Note: Dates in parentheses denote expiration of term.

Terms Expire November 30, 1927

G. M. Basford
Harte Cooke
E. R. Fish
O. P. Hood
Wm. F. Uhl

Terms Expire November 30, 1928

N. A. Carle
G. A. Goodenough
L. S. Marks
E. N. Trump
A. C. Wood

Terms Expire November 30, 1929

C. H. Berry
Francis Hodgkinson
D. S. Jacobus
L. F. Moody
E. B. Ricketts

SAFETY CODES

J. W. Upp, *Chairman and Representative on Council* (1926)
J. P. Jackson (1925)
C. B. Auel (1927)
H. L. Whittemore (1928)
G. E. Sanford (1929)

BOILER CODE

(Special Committee)

John A. Stevens, *Chairman*
D. S. Jacobus, *Vice-Chairman*
C. W. Obert, *Secretary*
Wm. H. Boehm
Frank H. Clark
F. S. Clark
Francis W. Dean
Thomas E. Durban
E. R. Fish
Elbert C. Fisher
Vincent M. Frost
Chas. E. Gorton
Arthur M. Greene, Jr.
Chas. L. Huston
S. F. Jeter
J. O. Leech
Fred R. Low
W. F. MacGregor
Edward F. Miller
M. F. Moore
I. E. Moultrop
C. O. Myers
James Partington
William B. Reed
C. L. Warwick
H. LeRoy Whitney

JOINT ACTIVITIES

IN WHICH THE SOCIETY FORMS A CORPORATE PART

The President and Secretary represent the Society on the Founder Societies' Joint Conference Committee, which directs all matters of joint interest into the proper channels.

AMERICAN ENGINEERING COUNCIL

Comprised of 28 member organizations to represent the engineers of America in matters of public welfare to the engineering and allied technical professions.

Terms Expire January, 1926

F. K. Copeland, Chicago
J. T. Faig, Cincinnati
R. E. Flanders, Springfield, Vt.
Dexter S. Kimball, Ithaca, N. Y.
W. B. Powell, Buffalo
Wm. Schwanhausser, New York
S. W. Stratton, Cambridge, Mass.
C. C. Thomas, Los Angeles, Cal.
P. F. Walker, Lawrence, Kan.
W. F. Durand, New York, *Chairman*

Note: Dates in parentheses denote expiration of term.

Terms Expire January, 1927

FRED R. LOW, New York, *Vice-Chairman*
IRA DYE, Seattle, Wash.
W. S. FINLAY, JR., New York
DEAN E. FOSTER, Tulsa, Okla.
WILSON P. HUNT, Moline, Ill.
I. E. MOULTROP, Boston
E. N. TRUMP, Syracuse, N. Y.
W. W. VARNEY, Baltimore, Md.

AMERICAN ENGINEERING STANDARDS COMMITTEE:
FRED E. ROGERS (1925)
STANLEY G. FLAGG, JR. (1926)
CLOYD M. CHAPMAN (1927)

EMPLOYMENT SERVICE BUREAU:
CALVIN W. RICE, Secretary A.S.M.E., *Chairman*

ENGINEERING FOUNDATION BOARD:
GEO. A. ORROK (1927)
A. M. GREENE, JR. (1928)

JOHN FRITZ MEDAL BOARD OF AWARD:
W. F. M. GOSS (February, 1926)
FRED J. MILLER (February, 1927)
HENRY B. SARGENT (February, 1928)
FRED R. LOW (February, 1929)

LIBRARY BOARD. See Library Committee, p. 11.

UNITED ENGINEERING SOCIETY:
W. F. M. GOSS (1926)
ROY V. WRIGHT (1927)
W. L. SAUNDERS (1928)

SOCIETY REPRESENTATION

OTHER ORGANIZATIONS IN WHICH THE SOCIETY IS REPRESENTED BY COURTESY

AMERICAN ASSOCIATION FOR THE ADVANCEMENT OF SCIENCE, SECTION M, ENGINEERING:
ALEX. C. HUMPHREYS
IRA N. HOLLIS

JOSEPH A. HOLMES MEMORIAL BOARD:
BRIGADIER GENERAL WM. A. BIXBY

MANAGEMENT WEEK COMMITTEE:
W. L. CONRAD

NATIONAL RESEARCH COUNCIL, DIVISION OF ENGINEERING:
ALBERT KINGSBURY (June, 1925)
R. J. S. PIGOTT (June, 1926)
GEO. A. ORROK (June, 1927)
To be appointed (June, 1928)

OIL AND GAS POWER WEEK COMMITTEE:
W. F. DURAND

NOTE: Dates in parentheses denote expiration of term.

SOCIETY FOR THE PROMOTION OF ENGINEERING EDUCATION, BOARD OF COÖRDINATION AND INVESTIGATION:
JOHN LYLE HARRINGTON
FRANK A. SCOTT

WESTERN SOCIETY OF ENGINEERS, WASHINGTON AWARD:
HERBERT S. PHILBRICK (June, 1925)
CHARLES RUSS RICHARDS (June, 1926)

NOTE: Dates in parentheses denote expiration of term.

SUMMARY OF MEMBERSHIP BY RESIDENCE

United States and Possessions

Alabama	109	Nebraska	30
Alaska	1	Nevada	3
Arizona	17	New Hampshire	41
Arkansas	12	New Jersey	1151
California	687	New Mexico	3
Canal Zone	5	New York	4420
Colorado	84	North Carolina	91
Connecticut	633	North Dakota	4
Delaware	76	Ohio	1085
District of Columbia	177	Oklahoma	76
Florida	95	Oregon	69
Georgia	122	Pennsylvania	1906
Hawaiian Islands	19	Philippine Islands	26
Idaho	3	Porto Rico	24
Illinois	1054	Rhode Island	160
Indiana	259	South Carolina	33
Iowa	42	South Dakota	7
Kansas	54	Tennessee	95
Kentucky	58	Texas	143
Louisiana	108	Utah	26
Maine	42	Vermont	35
Maryland	187	Virginia	109
Massachusetts	1146	Washington	109
Michigan	518	West Virginia	49
Minnesota	111	Wisconsin	322
Mississippi	12	Wyoming	9
Missouri	308		
Montana	25	Total	15989

Other Countries

Africa		**Asia**	
Egypt	2	China	22
Senegal	1	Dutch East Indies	1
Union of S. Africa	12	India	30
	15	Japan	26
Australasia		Manchuria	2
Australia	19	Persia	1
New Zealand	2	Siam	1
Tasmania	1	Straits Settlements	1
	22		84

Central America		
Costa Rica	4	
Guatemala	1	
Honduras	2	
		7
Europe		
Austria	2	
Belgium	4	
Czechoslovakia	5	
Denmark	7	
Finland	3	
France	33	
Germany	23	
Great Britain	88	
Greece	2	
Holland	2	
Italy	4	
Norway	3	
Poland	3	
Roumania	1	
Russia	1	
Spain	7	
Sweden	12	
Switzerland	8	
		211
North America		
Canada	229	
Newfoundland	4	
Mexico	34	
		267
South America		
Argentina	17	
Brazil	17	
Chile	22	
Colombia	4	
Peru	5	
Uruguay	1	
Venezuela	3	
		70
West Indies		
Cuba	68	
Dominican Republic	7	
Jamaica	2	
Trinidad	1	
		78
Total		751

Membership in United States	15989
Membership in Other Countries	751
Present Address Unknown	9
Total Membership	16749

Summary of Membership by Grades

Honorary Members	21
Members	7902
Associates	652
Associate-Members	3849
Juniors	4325
Total	16749

REPORTS OF GENERAL MEETINGS

ONLY the Spring and Annual Meetings of the Society are reported in TRANSACTIONS, and these not in detail. More complete accounts of the general meetings, and reports of the regional and sectional meetings were published during the year in *Mechanical Engineering* and the *A.S.M.E. News*.

THE SPRING MEETING

Milwaukee, Wis., May 18 through 21

The engineers of Milwaukee were effective hosts to a large group of engineers and operating men during the week beginning May 18, 1925. In addition to the Spring Meetings of The American Society of Mechanical Engineers and the American Society of Refrigerating Engineers, there was a gathering of the Wisconsin members of the National Association of Stationary Engineers and the Midwest Power Show held in the Auditorium.

The Spring Meeting of the A.S.M.E. was held from May 18 through 21 with headquarters at the Hotel Pfister. The registration was 1127, the largest in several years; the ladies scored a record attendance for any Spring Meeting, 118 being present.

The arrangements at Milwaukee were planned and conducted by able committees under the leadership of the following Executive Committee: Fred H. Dorner, *Chairman*, Robert Cramer, *Secretary*, Arthur Simon, *Treasurer*, W. C. Lindemann, A. C. Flory C. A. Cahill, J. D. Maurer, W. J. Sando, and E. T. John.

One of the outstanding events of the meeting was the banquet on Wednesday evening, May 20, which was addressed by Dwight F. Davis, Assistant Secretary of War, on the subject of Industrial Preparedness as Insurance against War. Fred H. Dorner, Chairman of the Milwaukee Committee, acted as toastmaster. Ex-Governor Emanuel L. Philipp of Wisconsin, in his remarks of welcome, paid tribute to the industrial leadership which had made Milwaukee a great machine-building center.

In response to Assistant Secretary Davis' address, Dr. William F. Durand, as President of the A.S.M.E., pointed out that the National Defense Division of the Society is working to develop means whereby the A.S.M.E. may fit itself effectively into the details of the problems of industrial preparedness.

As a token of esteem the engineers of Milwaukee presented to Dr. Durand at this dinner a painting of an attractive beach scene by Spicuzza, a Milwaukee artist.

In addition to the banquet the social program included an informal reception on Monday evening at which the mayor of Milwaukee greeted the guests. This was followed by dancing. The ladies held card parties and teas Tuesday and Thursday afternoons, and attended a musicale at the Art Institute Tuesday evening.

The excursions and plant visits at the Milwaukee meeting were very well planned and conducted. On Monday afternoon the visits included a trip to the sewage-disposal plant, the Palmolive Company, and the Phoenix Hosiery Company. On Tuesday afternoon about four hundred members boarded special cars for the Allis-Chalmers plant where luncheon was served. Careful routes had been marked throughout the extensive works and interesting exhibits thoroughly placarded had been moved to the inspection route so that in the time available the visitors had an excellent opportunity to become fully acquainted with the methods of manufacture and the products shown. On the same afternoon other groups visited Eline's chocolate factory, the plant of the Robert A. Johnston Company, and the Milwaukee Vocational School. On Wednesday afternoon excursions were made to the Riverside Pumping Station and the plants of the Vilter Manufacturing Company, the Falk Corporation, and the Kearney & Trecker Company. On Thursday afternoon 350 members were guests at luncheon at the plant of the Nordberg Manufacturing Company. After an inspection of the plant the party was taken to the Lakeside Power Station to see the method of burning pulverized fuel in successful operation there. Special excursions for the ladies included a visit to the public museum and an automobile sightseeing trip.

At the Business Meeting Monday afternoon the selection of San Francisco as the place for the 1926 Spring Meeting was announced, and the following codes were read by title: Proposed Standards for Steel Flanges and Flanged Fittings for Maximum Pressures of 250, 400, 600 and 900 Lb.; Proposed Standards for Cast-Iron Screwed Fittings and Maximum Pressures of 125 and 250 Lb.; Proposed Standards for Malleable-Iron Screwed Fittings for Maximum Pressure of 150 Lb.; Proposed Code for Identification of Piping Systems; and Proposed Safety Code for Elevators, Dumbwaiters, and Escalators.

There were several important committee meetings held during the meeting, among them being the Main Committee on Power Test Codes, the Boiler Code Committee jointly with the National Board of Boiler and Pressure Vessel Inspectors, and the Finance Committee with the Chairmen of Administrative Committees. The Special Research Committee on Metal Springs held an extended conference with a thorough discussion of the fundamental principles to be followed in the research program. The Council of the Society held its meeting on Monday.

Public Hearings

Three public hearings were held during the meeting. The first was on Monday afternoon under the auspices of the Boiler Code Committee when the suggested rules for the care of power boilers

were presented for public discussion. F. M. Gibson, Chairman of the Special Committee on the Care of Power Boilers, presided. On Wednesday afternoon the Test Code for Centrifugal and Rotary Pumps and the Test Code for Reciprocating Steam-Driven Displacement Pumps were presented at a public hearing with Fred R. Low, Chairman of the Main Committee on Power Test Codes, as presiding officer. Mr. Low also presided at the public hearing held jointly with the American Society of Refrigerating Engineers on Thursday afternoon, at which the Test Code for Refrigerating Systems was presented.

Technical Sessions

The papers for the Milwaukee meeting were printed well in advance, either in the special supplement to the May issue of *Mechanical Engineering* or in pamphlet form for distribution. The careful consideration given to the papers was reflected in the discussion they elicited, and the interest taken in the technical sessions was developed by this thorough preparation of printed matter for the meeting. The various sessions held and the papers presented are listed in the following summary of the technical program of the meeting.

Tuesday Morning, May 19

Machine-Shop Practice Session
(Under the auspices of the Machine-Shop Practice Division)

Recent Investigations in Turning and Planing and a New Form of Cutting Tool, Hans Klopstock.

Defects in Large Forgings, J. Fletcher Harper.

A Code of Design for Mechanical Springs, Joseph K. Wood.

Hydroelectric Session
(Under the auspices of the Power Division)

Mechanical Problems of Hydraulic-Turbine Design, William Monroe White.

The Parallel Operation of Hydro and Steam Plants, F. A. Allner.

Mechanical Features Affecting the Reliable and Economical Operation of Hydroelectric Plants, E. A. Dow.

Forest Products Session
(Under the auspices of the Forest Products Division)

A New Era in Forestry, Howard F. Weiss.

Standardization of Grading Rules for Hardwood Lumber, A. T. Upson.

Material Handling in a Piano Factory, P. H. Bilhuber.

Discussion of Government plan for lumber conservation as stated in Clark-McNary Bill, and methods of reducing waste in wood-using factories.

Pulverized Coal Session
(Under the auspices of the Fuels Division)

Boiler Furnaces for Pulverized Coal, A. G. Christie.

Radiation in the Pulverized-Fuel Furnace, Walter J. Wohlenberg and Donald G. Morrow.

A Microscopic Study of Pulverized Coal, L. V. Andrews.

Tuesday Evening, May 19

MILWAUKEE SESSION
(Under the auspices of the Milwaukee Committee)

The Activated-Sludge Sewage-Disposal Plant at Milwaukee, JOHN ARTHUR WILSON.

Critical Study of Heat and Power Requirements of Sewage-Disposal Plants, ROBERT CRAMER.

Duty Tests of Vertical Triple-Expansion Pumping Engines, Milwaukee, Wis., CHARLES A. CAHILL.

The Economical Advantages of Cities Having Diversified Industries, FRED H. DORNER.

Wednesday Morning, May 20

MATERIALS SESSION

The X-Ray Examination of Steel Castings, I. E. MOULTROP and E. W. NORRIS.

Aluminum and Its Light Alloys, ROBERT L. STREETER and P. V. FARAGHER.

Stress Concentration Produced by Holes and Fillets, S. TIMOSHENKO and W. DIETZ.

MATERIALS-HANDLING SESSION
(Under the auspices of the Materials-Handling Division)

Formulas for Computing the Economies of Labor-Saving Equipment, JAMES A. SHEPARD and GEORGE E. HAGEMANN.

An Application of the Formulas for Computing Economies of Labor-Saving Equipment, GEORGE LANGFORD, JR.

Labor-Saving Equipment in Road Construction, E. H. LICHTENBERG and JAMES A. SHEPARD.

Economic Efficiency of the Full-Automatic Turret Lathe in Comparison with the Semi-Turret Lathe, R. J. WADD.

NATIONAL DEFENSE SESSION
(Under the auspices of the National Defense Division and Chicago Procurement Planning Association)

Discussion of Procurement Planning and Industrial Mobilization. Leaders: A. M. SHEARER, E. A. RUSSELL, R. P. LAMONT, COL. C. L. HINKLE, and HOBART S. JOHNSON.

INDUSTRIAL POWER SESSION
(Under the auspices of the Oil and Gas Power Division)

Torsional Vibrations and Critical Speeds of Shafts, ARNOLD LACK and CHARLES B. JAHNKE.

Test of a Uniflow Engine, GEORGE H. BARRUS.

Thursday Morning, May 21

STEAM POWER SESSION
(Under the auspices of the Power Division)

Lake Waters for Condensers, A. G. CHRISTIE.

The Rational Design of Covering for Pipes Carrying Steam up to 800 Deg. Fahr., W. A. CARTER and E. T. COPE.

A Review of Steam-Turbine Development, HANS DAHLSTRAND.

Analysis of Power-Plant Performance Based on the Second Law of Thermodynamics, WILLIAM L. DEBAUFRE.

Comparison of Actual Performance and Theoretical Possibilities of the Lakeside Station, M. K. DREWRY.

APPRENTICESHIP SESSION
(Under the auspices of the Committee on Education and Training for the Industries)

A National Apprenticeship Program, HAROLD S. FALK.
Discussion: The District Apprenticeship Plan of Milwaukee, H. S. ROGERS.
Discussion: The Place of Railroad Apprenticeships in a National Apprenticeship Plan, F. W. THOMAS.
Discussion: The Place of Vocational Schools in the District Apprenticeship Plan, R. L. COOLEY.
Presentation of District Apprenticeship Plan as Practiced in Milwaukee, H. A. FROMMELT and JOHN J. METZ.

MANAGEMENT SESSION
(Under the auspices of the Management Division)

Steel Foundry Management, R. A. BULL.
Management of Gray-Iron Foundries from Management and Cost Standpoint, G. P. FISHER.
Report of Committee on Elimination of Unnecessary Fatigue in Industry, Presented by GEORGE H. SHEPARD.

RAILROAD SESSION
(Under the auspices of the Railroad Division)

Factors Concerning the Economics of Shopping Steam Locomotives, L. K. SILLCOX.
Some Freight-Car Maintenance Problems, CHAS. G. JUNEAU.

THE ANNUAL MEETING

New York, November 30 through December 4

Two cabinet officers, two outstanding scientists, and over sixty leaders of engineering and industrial thought participated in the program of the Forty-Sixth Annual Meeting. The registration of 2027 places the meeting among the four largest the Society has ever held.

The features of greatest general interest included the Henry Robinson Towne Lecture by the Honorable Herbert Hoover, the presidential address by Dr. William F. Durand, the annual dinner with Dr. M. I. Pupin as the principal speaker, the presentation of a splendid collection of progress reports by the Professional Divisions, the Robert Henry Thurston Lecture by Dr. Zay Jeffries, the all-day visit to the Bethlehem Steel Company at Bethlehem, Pa., and the session on Industrial Coöperation with the War Department which was presided over by Judge E. H. Gary and addressed by Secretary of War Dwight F. Davis, Assistant Secretary of War Hanford MacNider, Gen. James G. Harbord, and Gen. C. P. Summerall. An important factor in the success of the technical sessions was the large amount of good discussion that was brought out. In the same way as for the Spring Meeting, this was made possible by the comprehensive plan for printing and distributing papers in advance so that the members could have opportunity to study them and prepare discussions.

The meeting included the usual coöperating sessions with the American Society of Refrigerating Engineers and the Taylor Society, both of which hold their annual meetings during the week of the A.S.M.E. meeting. The session with the A.S.R.E. was devoted to the subject of centrifugal compressors. The Management Division of the A.S.M.E. conducted two sessions jointly with the Taylor Society. In addition, the Taylor Society held a well-attended meeting on Thursday evening, December 3, which was addressed by William Green, president of the American Federation of Labor, on Labor's Ideals Concerning Management.

Presidential Address and Award of Honorary Memberships

An impressive audience filled the auditorium of the Engineering Societies Building on Tuesday evening to hear the Presidential Address, to witness the award of Honorary Memberships to two distinguished engineers, and to greet the incoming President of the Society.

Dr. Durand's address as retiring president was a presentation of the engineer's growth with civilization and his responsibility for the future. It forms a valuable addition to the literature of the profession.

Simple but impressive ceremonies marked the award of Honorary Memberships to Worcester Reed Warner and Herbert Clark Hoover. Mr. Warner was presented to President Durand by Dr. Alexander Humphreys, Past-President of the Society, who briefly related his achievements in astronomical science, in industry, and as a citizen. Mr. Warner responded gratefully to this recognition by the Society, in whose work he had actively participated since its founding. Dr. Fred R. Low, Past-President of the Society, introduced Mr. Hoover to the President as an engineer-statesman who has demonstrated the benefits that may be derived from applying the engineering way of thought and analysis to the problems of government. Mr. Hoover expressed his deep appreciation of the honor tendered him by his fellow-engineers, saying that the true test of a man was his standing with his professional brethren.

The results of the letter ballot of the membership for the election of officers were announced by David B. Porter, chairman of the Board of Tellers. The following officers were declared elected:

President: William L. Abbott

Vice-Presidents: A. G. Christie, William T. Magruder, Roy V. Wright

Managers: Robert L. Daugherty, William Elmer, Charles E. Gorton

Representatives on American Engineering Council: William L. Abbott, John T. Faig, A. M. Greene, Jr., John L. Harrington, Ira N. Hollis, S. H. Libby, Dexter S. Kimball, William B. Powell, H. L. Thompson.

Mr. Abbott, the President-Elect, was escorted to the rostrum by Dr. D. S. Jacobus and Dr. Fred R. Low, Past-Presidents of the Society, and received the gavel from Dr. Durand with an expression of the latter's good wishes for his administration. After a brief acknowledgment of the honor by Mr. Abbott, the gathering adjourned to the fifth floor of the Engineering Societies Building, where the President, the President-Elect, and the two recipients of Honorary Membership were tendered a reception. Dancing followed.

The Towne and Thurston Lectures

Economics and science took their place in the program of the A.S.M.E. Annual Meeting with the institution of the Henry Robinson Towne and Robert Henry Thurston Lectures. If the function of the engineer is to translate the research results of the scientist into benefits for mankind, he will have relations with the scientific world on the one hand and the business world on the other. These relationships extend through wide zones of indefinite limits, and it is the purpose of the two lectures to give an opportunity for leaders of thought in these zones to present the latest developments to the A.S.M.E. The lecture on the relations between engineering and economics was entitled the Henry Robinson Towne Lecture in recognition of the service which Mr. Towne, a Past-President of the A.S.M.E., rendered in establishing the engineer in the field of management and economics. The science lecture was designated the Robert Henry Thurston Lecture in honor of Dr. Thurston, who, as the first President of the A.S.M.E. and as a great leader of engineering thought, was instrumental in developing mechanical engineering from a collection of empiricisms into a well-grounded science.

The Honorable Herbert Hoover, Secretary of Commerce, delivered the first Henry Robinson Towne Lecture on Tuesday afternoon, December 1. He took as his subject the Economic Value of Research in Pure Science. The first Robert Henry Thurston Lecture was delivered by Dr. Zay Jeffries, an outstanding metallurgist, on Thursday afternoon, December 3. The subject was Engineering and Science in the Metal Industry.

Industrial Coöperation with the War Department

A splendid gathering of engineers, chemists, and manufacturers crowded the Auditorium of the Engineering Societies Building on Friday evening, December 4, to greet the War Department heads and express wholehearted support of the plans for industrial preparedness developed under the National Defense Act of 1920.

The opening event was the presentation of the colors of the 24th Regiment of Engineers to the custody of the United Engineering Society. Lieutenant-Colonel Elliot H. Whitlock, of Cleveland,

made the presentation and W. L. Saunders, President of the United Engineering Society, received them. Judge Elbert H. Gary, Chairman of the New York Ordnance District Advisory Board, then took the chair. The program included addresses by the Honorable Dwight F. Davis, Secretary of War, the Honorable Hanford MacNider, Assistant Secretary of War, Gen. James G. Harbord, President of the Radio Corporation of America, and Gen. C. P. Summerall, Commanding-General of the Second Corps Area.

The Dinner

The annual dinner was held at the Hotel Astor on Wednesday evening, December 2. The speaking program was intensely interesting and short, allowing plenty of time for the informal foregathering of old acquaintances. A large number of ladies were present, and after the dinner the floor was cleared for dancing.

Samuel H. Libby won fame as toastmaster by his unique introduction of the speakers and the atmosphere of good cheer and professional fellowship that he maintained.

The first part of the program was devoted to the men who had joined the Society during the preceding year, each of whom had an opportunity of meeting the officers of the Society at an informal reception preceding the dinner. The distinguished guests and the past-presidents of the Society who occupied seats at the speakers' table included Robert Ridgeway, President of the American Society of Civil Engineers, J. V. W. Reynders, President of the American Institute of Mining and Metallurgical Engineers, Van R. H. Greene, President of the American Society of Refrigerating Engineers, Percy S. Brown, President of the Taylor Society, Past-Presidents Worcester R. Warner, Ambrose Swasey, Jesse M. Smith, D. S. Jacobus, Dexter S. Kimball, John Lyle Harrington, and Fred R. Low, and President-Elect W. L. Abbott. These men were introduced by Dr. Durand, following which Secretary Calvin W. Rice called the roll of 35 new members present, each of whom rose in his place. Past-President Low then addressed the new members, explaining the purpose of the Society, relating its accomplishments, and pointing out the opportunities that it offers for professional comradeships, for encouragement for greater accomplishment, and for broadening the individual's circle of living and opportunity for usefulness.

The principal speaker of the evening was Dr. Michael I. Pupin, President of the American Institute of Electrical Engineers and of the American Association for the Advancement of Science. By vivid descriptions and anecdotes he carried his audience to the primitive Serbian village where he was born and raised, to the Delaware farm where he learned to manage a pair of stubborn mules, and to the old factory on Chambers street in New York City where he saw for the first time a real steam engine in a real

boiler room. He spoke parables of engineering idealism based on these trying experiences of his early days in a manner that raised his audience to a new plane of appreciation of the dignity of the engineering profession.

The Social Program

In addition to the dinner, which was the most important social event, there was an informal get-together on Monday evening, November 30, on the fifth floor of the Engineering Societies Building, and various events conducted by the Woman's Auxiliary of the Society. These included a get-together on Monday night, with special music and cards, a visit to the International House where luncheon was served and tea at New York University on Tuesday, a tour of lower New York on Wednesday morning, the annual reception and tea Wednesday afternoon, and an informal reception and luncheon on Thursday at the Town Hall Club, at which the speaker was P. Whitwell Wilson, a former member of the British Parliament and a journalist of international standing.

Excursions

The feature excursion of the meeting was the all-day trip to Bethlehem, Pa., on Friday, December 4, to see the works of the Bethlehem Steel Corporation. Visits were also made to the De La Vergne Machine Company, the Hellgate Station of the United Electric Light and Power Company, the New York-New Jersey Vehicular Tunnel, the Hudson Avenue Station of the Brooklyn Edison Company, the yards of the Central Railroad of New Jersey to see the new 80-ton Diesel-electric locomotive, the Lee Spring Co., Steinway & Sons, the Edison Lighting Institute, and the Motorship *Gripsholm*.

Council Meetings

The Council met twice during the meeting, its first session being on Monday, November 30. The principal business was the election of John A. Stevens as Honorary Chairman of the Boiler Code Committee in Perpetuity. Mr. Stevens had recently resigned as chairman after fourteen years of active service, and this unusual honor by Council was in recognition of his great contribution of time and effort to the work of the Society.

The date of the Providence Meeting was decided upon as May 3 through 6, 1926. The remainder of the meeting was given over to routine business which lasted through the morning and afternoon.

On Friday, December 4, W. L. Abbott as the new President received the gavel of authority from Dr. Durand and the new members of the Council were installed. Calvin W. Rice was

reëlected Secretary. The following were elected as the Executive Committee of the Council: President W. L. Abbott, Past-Presidents Dexter S. Kimball, Fred R. Low and William F. Durand, Vice-President Roy V. Wright, and Manager John H. Lawrence. The remainder of the meeting was given over to the appointment of committee personnel for the ensuing year.

Local Sections' Delegates Conference

The conference of delegates from the Local Sections of the Society marked the tenth anniversary of the initiation of the Local Sections movement. The conference occupied all of Monday, November 30, and smaller groups of delegates were in session throughout the meeting. Each Section answered the roll call by giving a short review of the accomplishments of the year. There were also brief addresses by leaders of various Society activities. Under the discussion of Society meetings, the delegates expressed unanimous approval of White Sulphur Springs as the Spring Meeting place in 1927. The delegates of each Section on the route of the A.S.M.E. special train to the San Francisco Spring Meeting in 1926 gave assurance of a hearty welcome.

At noon the members of the Council joined the delegates at luncheon. Mr. Penrose, retiring chairman of the Committee on Local Sections, presided, and Dr. Durand and Mr. Abbott spoke of the splendid work of the Local Sections in advancing the work of the Society.

Student Branch Conference

Forty of the eighty-three student branches were represented at a Student Branch conference held Wednesday afternoon, December 2, following a luncheon meeting with members of the Council. The conference was taken up with the discussion of means of stimulating Student Branch activities and with the presentation of information regarding the work of the Society.

Business Meeting — Progress Reports

The Business Meeting of the Society, held on Wednesday afternoon, December 2, 1925, was given an unusual flavor by the inclusion of the presentation of reports on progress and prospects in mechanical engineering. The Professional Divisions had prepared a splendid collection of statements covering the field of mechanical engineering. These were printed in full in the December, 1925, issue of *Mechanical Engineering* and were presented in brief by the representatives of the Divisions. The combined reports will serve as the basis for future programs of meetings of the Society, and the selection of research and standardization projects.

During 1925 the Society made an investigation of mechanical-engineering education to assist in the broad, comprehensive investigation of engineering education being carried out by the Society for the Promotion of Engineering Education. The study conducted by the A.S.M.E. consisted of a questionnaire investigation of the membership of the A.S.M.E. to discover the facts concerning the education of the members of the Society and their views in regard to desirable characteristics of mechanical-engineering curricula. This study was conducted by John Lyle Harrington, Frank A. Scott, and W. L. Durand, who represent the A.S.M.E. on the Board of Investigation of the S.P.E.E. A brief report was presented by Mr. Harrington at the Business Meeting.

The bestowal of Society awards has always furnished an occasion of great interest. This year, Dr. Ira N. Hollis, Chairman of the Committee on Awards, opened the ceremony by a brief explanation of the value of the A.S.M.E. awards in developing the power of expression among young engineers. The recipients of the awards were then introduced to the President by Dr. Hollis. The Charles T. Main Award was made to Clement R. Brown, of Washington, D. C., for the best paper on The Influence of the Locomotive upon the Unity of Our Country. His paper appeared in the December, 1925, issue of *Mechanical Engineering*. The Junior Award was made to Gilbert S. Schaller, Assistant Professor of Engineering Shops, at the University of Washington, Seattle, for his Investigation of Seattle as a Location for a Synthetic Foundry Industry. In Mr. Schaller's absence the prize was given to Prof. E. O. Eastwood, of Seattle, to be bestowed upon Mr. Schaller at a meeting of the Western Washington Section of the A.S.M.E. A Student Award was made jointly to William S. Montgomery and E. Ray Enders, both of Pennsylvania State College, for their paper on Some Attempts to Measure the Drawing Properties of Metal. A second Student Award was made to Harry Pease Cox, Jr., for his paper on A Study of the Effect of End Shape on the Towing Resistance of Barge Models.

The routine matters brought up included the annual report of the Council, which was presented by the Secretary, and the reading of the following standards by title:

Safety Code for Elevators, Dumbwaiters, and Escalators (First Revision)

Cold-Finished Shafting, Standard Diameters and Lengths (Transmission and Machinery)

Square and Flat Stock Keys, Standard Widths and Heights.

The following recommendation of the Conference of Local Sections Delegates for the personnel of the Nominating Committee was presented by W. A. Hanley of the Committee on Local Sections and approved by vote of the meeting:

GROUP I — B. S. Lewis, New Britain, Conn. I. E. Moultrop, Boston, Mass., *Alternate*.

GROUP II — J. J. Nelis, New York City. L. B. McMillan, New York City, *Alternate*.

GROUP III — B. M. Brigman, Louisville, Ky. Chas. Loeber, Richmond, Va., *Alternate*. V. L. Sanderson, Philadelphia, *2d Alternate*.

GROUP IV — E. G. Bailey, Cleveland. C. P. Fortney, Charlestown, W. Va., *Alternate*.

GROUP V — C. F. Hirshfeld, Detroit. J. D. Cunningham, Chicago, Ill., *Alternate*.

GROUP VI — Jiles W. Haney, Lincoln, Neb. James M. Robert, New Orleans, La., *Alternate*.

GROUP VII — R. L. Rockwell, Seattle, Wash. H. L. Doolittle, Los Angeles, Cal., *Alternate*.

The group numbers refer to the geographical groups of the sections which select representation on the Nominating Committee. At a subsequent meeting the Nominating Committee selected C. F. Hirshfeld, of Detroit, Mich., as chairman, and B. M. Brigman, of Louisville, Ky., as secretary.

Following the routine business, John C. Parker, of Philadelphia, Pa., spoke suggesting the encouragement of the discussion of Society policies at the Business Meeting, and Luther D. Burlingame of Providence, R. I., extended a hearty welcome to the Society to the New England Regional Meeting to be held in Providence May 3 to 7, 1926.

TECHNICAL SESSIONS

A public hearing was held on Tuesday morning, December 1, at which the Power Test Code for Steam Turbines was presented for criticism. C. Harold Berry, as Chairman of the Committee on the Power Test Code for Steam Turbines, presided. The technical sessions held Tuesday, Wednesday, and Thursday were as follows:

Tuesday Morning, December 1

OIL AND GAS POWER SESSION
(Under auspices of Oil and Gas Power Division)

Centrifugal Compressors for Diesel Engines, S. A. Moss.

Auxiliaries for Motor Vessels, JOHN W. MORTON and A. B. NEWELL.

Presentation of Award for Best Paper Delivered during Oil and Gas Power Week, 1925, to COM. E. E. WILSON, Bureau of Aeronautics, U. S. N.

Electric Transmission for Internal Combustion Engines, HERMANN LEMP.

The Gas Engines of the Maryland Plant of the Bethlehem Steel Company, A. A. RAYMOND.

MACHINE-SHOP PRACTICE (SESSION I)
(Under auspices of Machine-Shop Practice Division)

The Tension Ratio and Transmissive Power of Belts, C. A. NORMAN.

Principles and Advantages of Optical Methods for Measuring Machine Parts, HENRY F. KURTZ.

WOOD INDUSTRIES SESSION
(Under auspices of Wood Industries Division)

The Electric Molder, ANDREW JENSEN, JR.
Scientifically Developed Shipping Containers, C. M. BONNELL, JR.
High-Speed Induction Motors and Frequency Changers, CHARLES FAIR.
Recent Advances in Methods of Glue Evaluation, WILBUR L. JONES.
Spark Arresters and Forest Fires, J. S. MATHEWSON.

Tuesday Afternoon, December 1

SESSION ON INDUSTRIAL POWER
(Under auspices of Power Division)

The Value of Higher Steam Pressures in the Industrial Plant, WILLIAM F. RYAN.
The Supply of Industrial Power, WM. HARRISON LARKIN, JR.

SESSION ON CENTRIFUGAL COMPRESSORS
(Joint Session with American Society of Refrigerating Engineers)

The Heat Balance Method of Testing Centrifugal Compressors, M. G. ROBINSON.
Centrifugal Compression as Applied to Refrigeration, W. H. CARRIER.

SESSION ON CALCULATION METHODS

Graphical Methods of Calculation, HERBERT L. SEWARD.
The Making of Special Slide Rules, G. W. GREENWOOD.

Wednesday Morning, December 2

SESSION ON INDUSTRIAL FURNACES
(Under auspices of Fuels Division)

Industrial Furnaces, W. TRINKS.
Industrial-Furnace Efficiency, VICTOR J. AZBE.

SESSION ON MATERIALS HANDLING
(Under auspices of Materials Handling Division and Safety Committee)

Materials-Handling Problems and Their Solution, F. D. CAMPBELL.
Safety in Materials Handling, DAVID S. BEYER.
Progress Report on Application of Materials Handling Formulas, JAMES A. SHEPARD.

MACHINE-SHOP PRACTICE (SESSION II)
(Under auspices of Machine-Shop Practice Division)

Some Comparative Wear Experiments on Cast-Iron Gear Teeth, G. H. MARX, L. E. CUTTER and B. M. GREEN.
Normal Pitch—The Index of Gear Performance, G. M. EATON.
The Question Mark in Machine Design, FORREST E. CARDULLO.

SESSION ON SPRINGS
(Under auspices of Special Research Committee on Metal Springs)

Phosphor-Bronze Helical Springs from the Standpoint of Precision Instruments, W. G. BROMBACHER.
The Manufacture of Commercial Steel Helical Springs, F. H. BROWN.
Characteristics of Weighing Springs, J. W. ROCKEFELLER, JR.
Springs for Electrical Measuring Instruments, B. W. ST. CLAIR.
Formulas for the Design of Helical Springs of Square or Rectangular Steel, C. T. EDGERTON.
An Outline for the Application of Fatigue and Elastic Results to Metal Spring Design, T. MCLEAN JASPER.
The Ring Spring, O. R. WIKANDER.

Wednesday Afternoon, December 2

SESSION ON EDUCATION AND TRAINING FOR THE INDUSTRIES OF NON-COLLEGE TYPE

(Under auspices of Committee on Education and Training for the Industries)

Education for the Industries of Non-College Type in Europe, WILLIAM E. WICKENDEN.

The Apprenticeship Movement and Its Relation to the Building Trades, DWIGHT L. HOOPINGARNER.

STEAM TABLES RESEARCH

Report of Executive Committee, Steam Table Fund, GEO. A. ORROK.

Research Progress Reports, F. G. KEYES and N. S. OSBORNE.

Progress Report on the Development of Steam Charts and Tables from the Harvard Throttling Experiments, J. H. KEENAN.

Summation of Research Results, HARVEY N. DAVIS.

Thursday Morning, December 3

STEAM POWER SESSION

(Under auspices of Power Division)

Recent Developments at Colfax Station, Duquesne Light Co., CHAS. W. E. CLARKE.

Steam Bleeding and Turbine Performance, C. D. ZIMMERMAN.

Radiation in Boiler Furnaces, B. N. BROIDO.

The Development of a Unit Pulverizer, R. SANFORD RILEY and OLLISON CRAIG.

MANAGEMENT SESSION

(Under auspices of Management Division and Taylor Society)

Influence of Plant Design on Plant Efficiency, HAROLD T. MOORE.

Production Control in the Newsprint Industry, GEORGE D. BEARCE.

Discussion on the Teaching of Management Engineering.

Carbon Dioxide as an Index of Fatigue, WALTER N. POLAKOV.

SESSION ON AERONAUTICS

(Under auspices of Aeronautic Division)

Technical Progress at McCook Field, LIEUT. E. E. ALDRIN.

The Airship and Its Place in Commerce, LIEUT. COLONEL H. H. BLEE.

Maintenance and Depreciation of Airplanes and Engines, LIEUT. ERNEST W. DICHMAN.

SESSION ON TEXTILES

(Under auspices of Textile Division)

Individual Motor Application to Woolen and Cotton Card Machinery, WM. A. MAYOR.

Power for Textile Mills, CHAS. T. MAIN.

Thursday Afternoon, December 3

SESSION ON POWER PLANT MATERIALS

(Under auspices of Power Division)

Heat-Treatment Data on Quality Steel Castings, ALBERT E. WHITE.

Bolts for Use in Power-Plant Construction, WILLIAM P. WOOD.

Boiler-Furnace Refractories, E. B. POWELL.

SESSION ON INDUSTRIAL PSYCHOLOGY

(Under auspices of Management Division and Taylor Society)

The Present State of Industrial Psychology, LILLIAN M. GILBRETH.

Psychology in the Organization of Prison Industries, EDGAR A. DOLL.

SESSION ON LUBRICATION

(Under auspices of Special Research Committee on Lubrication)

Graphical Study of Journal Lubrication (Part III), H. A. S. HOWARTH.

Charts for Studying the Oil Film in Bearings, GEORGE B. KARELITZ.

SESSION ON DESIGN

Torsional-Stress Concentrations in Shafts of Circular Cross-Section and Variable Diameter, L. S. JACOBSEN.

Tangential Vibration of Steam-Turbine Buckets, WILFRED CAMPBELL and W. C. HECKMAN.

Vibration Phenomena of a Loaded Unbalanced Shaft while Passing through Its Critical Speed, A. L. KIMBALL, JR., and E. H. HULL.

At the request of President Durand, there was read at each session of the Annual Meeting on Thursday morning the following statement presenting a critical situation in the industrial and engineering life of our country. Each session was declared suspended for a period of one minute for the quiet consideration of the gravity of the accident losses with which the country is confronted, and the acceptance of the pledge outlined was indicated by those in attendance remaining silent for that period.

The accident situation in this country has reached appalling proportions, taking an annual toll of approximately 80,000 lives, while disabling over 2,000,000 more for varying periods. This yearly loss is greater than that sustained by our armies during the entire period of the World War, and has become still more serious in several of its aspects.

Compared with other nations, there are killed in peaceful America per million of population almost twice as many as in Great Britain, and four times as many as in Denmark.

The direct loss has been estimated to run each year into the billions of dollars, while the indirect loss is beyond calculation, and our country is now confronted with a problem, already sufficiently grave, which may ultimately prove to be the greatest in its history.

Additional to and dwarfing all such losses, however, are the human suffering and misery engendered, the extent of which none can either measure or conceive, but which must exert an increasingly retarding effect on the advancement and uplift of our country and bring a stain upon it which can never be effaced.

The members of this Society, therefore, in recognition of the situation, which calls for the best efforts of every citizen having the welfare of our country at heart, pledge themselves and this Society to continued and unremitting effort in this greatest of all human endeavor — the work of accident prevention.

The Annual Meeting Committees were under the general chairmanship of Emmett B. Carter. Their chairmen were as follows: L. F. Lyne, Jr., *Reception;* A. J. Sicree, *Information;* Geo. F. Felker, *Courtesy;* L. H. Welling, *Excursions;* V. M. Frost, *Open House;* Clyde R. Place, *President's Reception;* Fred R. Low, *Dinner;* Warren D. Lewis, *Catering.* Mrs. R. V. Wright was general chairman for the ladies' committees, and the following were chairmen of the various sub-committees: Mrs. C. B. LePage, *Get-Together;* Mrs. L. R. Gurley, *Information and Excursions;* Mrs. R. M. Gates, *Hospitality;* and Mrs. G. L. Knight, *Luncheon.*

No. 1953

A CODE OF DESIGN FOR MECHANICAL SPRINGS

By Joseph Kaye Wood,[1] New York, N. Y.

Associate-Member of the Society

This paper first gives an entirely new derivation of the general spring formulas developed by the author from the numerous scattered orthodox spring formulas and presented in his paper on Mechanical Springs at the A.S.M.E. Annual Meeting in December, 1924. The two sets of formulas, although derived independently, are identical in fabrication of terms, and since the new derivation is based upon the direct application of the individual terms, this agreement establishes the general formulas on a more logical and practical basis. There are some differences, however, in the specific values of the stress-method and form-of-section constants, which in the case of springs of the uniform-flexure type account exactly for differences which have been found between calculated and actual test results. Complete research is urged to confirm this finding definitely, and until such work is done the author will employ the constants that agree with the orthodox formulas. In view of the remarkable manner in which the new derivation establishes the logical arrangement of the generalized formulas, the author has drawn up and included in the paper a brief code of design for mechanical springs, which he hopes will serve as a nucleus for a more complete and final code in the future.

AT THE Annual Meeting of The American Society of Mechanical Engineers in December, 1924, the author presented a paper[2] on Mechanical Springs, in which a general method of design for springs was described. In the summary, it was stated that this method possibly could provide a good foundation upon which to base a suitable code for spring design, because of its conciseness, logical arrangement, simplicity, and the manner in which it groups the otherwise greatly diversified and complex methods of design.

[1] Consulting Engineer.

[2] See Trans., A.S.M.E., vol. 46 (1924), p. 915.

Contributed by the Machine Shop Practice Division and presented at the Spring Meeting, Milwaukee, Wis., May 18 to 21, 1925, of The American Society of Mechanical Engineers.

2 Since that time further study of the general method has led the author to the discovery of a general derivation for all spring formulas, which makes the method so logical as to justify its use as a basis for codification. However, since the art of spring design has not been developed or standardized to a sufficient degree to warrant final codification, the present paper is confined to the exposition of the general method of spring design in codified form. Preceding the presentation of this tentative code are the derivations of the general spring formulas which show the significance of each term in the formulas. Another important result of this discovery is the possible explanation of a certain discrepancy that is known to exist in formulas for springs of the flexure type.

3 The development of such a code, although tentative and unofficial, can be made, therefore, to serve two very useful purposes: namely, to provide a clearer vision as to the ultimate or final code desired, and in the meantime to enable the design of mechanical springs to be accomplished in a more efficient manner than is possible in the present haphazard state of the art.

4 The paper referred to gave the derivation of the following formulas, which, while in accordance with admitted principles of elasticity, are expressed in such a form as to be applicable to mechanical springs in general.

Load-Deflection Rate:

$$\frac{P}{F} = kE\,\frac{A}{l}\left(\frac{d}{D}\right)^z \quad \ldots\ldots\ldots [1]$$

Safe Maximum Load:

$$P_m = mSA\left(\frac{d}{D}\right) \quad \ldots\ldots\ldots [2]$$

Safe Maximum Deflection:

$$F_m = n\,\frac{S}{E}\,l\left(\frac{D}{d}\right) \quad \ldots\ldots\ldots [3]$$

Safe Maximum Work:

$$W_m = \frac{u}{2}\,\frac{S^2}{E}\,Al \quad \ldots\ldots\ldots [4]$$

In the above formulas

$\frac{P}{F}$ = load-deflection rate, lb. per in.

P_m = load, safe maximum, lb.

F_m = deflection, safe maximum, in.

W_m = work, safe maximum, in-lb.

$\frac{S^2}{E}$ = modulus of resilience, in-lb. per cu. in.

$\frac{S}{E}$ = material index, safe working (pure number)

S = unit tensile stress, safe maximum, lb. per sq. in. (= 1.6S for torsion)

E = tensile modulus of elasticity, lb. per sq. in. ($= 2.5G$, where G = modulus for torsion)
Al = volume of bar, cu. in.
$\frac{A}{l}$ = bar ratio, in.
A = area of bar cross-section, sq. in.
l = length of bar, in.
$\left(\frac{D}{d}\right)$ = lever index (pure number)
D = major lever arm or moment arm of load, in.
d = minor lever arm (twice maximum), in.

Stress-Method and Form-of Section Constants

k = load-deflection-rate constant
m = safe-maximum-load constant
n = safe-maximum-deflection constant
u = safe-maximum-work constant.

5 The above terms are drawn from the many types of widely scattered spring formulas, derived, as stated previously, in accordance with admitted principles of elasticity and then incorporated into the four simple generalized spring formulas shown. The significance of these terms and their general arrangement are illustrated by the following new derivations which are more direct and general.

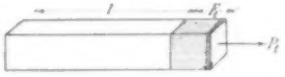

FIG. 1

NEW DERIVATIONS OF GENERAL SPRING FORMULAS

6 Let it be assumed for the moment that only one method of stressing a bar is known, namely, that of pure tension. The bar shown in Fig. 1, l in. long and of uniform cross-sectional area A sq. in., is stretched F_t in. by a force of P_t lb. The resulting unit stress S in lb. per sq. in. is equal to P_t/A, while the load-deflection rate P_t/F_t in lb. per in. is evaluated as follows:

$$\frac{P_t}{F_t} = E\frac{A}{l} \quad \ldots\ldots\ldots\ldots [5]$$

where A/l is the bar ratio and E is the tensile modulus of elasticity.

7 In Equation [5] the bar ratio might be considered as an operator which modifies the impractically high value of E (the lowest value of E for metals is equal to 6,000,000) so as to give a much lower load-deflection rate. Suppose, for example, a rate of 1000 lb. per in. is desired for a metal having a value of E of about 6,000,000 lb. per sq. in. The bar ratio would need to be equal to 1/6000, and if a maximum deflection of 1 in. is specified, the area

component of this ratio for a given elastic limit of 30,000 lb. per sq. in. would need to be 0.033 sq. in. This would limit the length l in the bar ratio to a minimum of 200 in., which in a straight tensile piece would be impractical for designing purposes, as space limitation is of importance. Reducing the length to the desired value increases the load-deflection rate to a prohibitive amount. In other words, with the resilient energy obtained from a reasonably short length of bar having a moderately small cross-sectional area, the load component is much too large and the deflection component much too small to give a load-deflection rate of a value suitable for use in mechanical design. The question arises, therefore, that with the limitations imposed by the requirements of mechanical design, how can the deflection component be increased relatively to its load component, or in other words, how can the load-deflection rate be decreased?

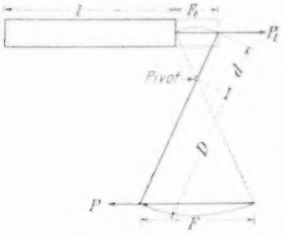

FIG. 2

8 Since the maximum resilient energy is

$$W_m = \frac{S^2}{2E} Al \quad . \quad . \quad . \quad [6]$$

and since for a given material the volume Al cannot be varied sufficiently to give the desired result, some external means must be employed. Fig. 2 shows an external device in the form of a mechanical lever whose arms are d and D. By applying a force P to the end of arm D in the direction shown and moving it through a distance F, the bar of length l and cross-sectional area A is stretched an amount equal to F_t. Then

$$P_t = \frac{D}{d} P$$

and

$$F_t = \frac{d}{D} F$$

Combining,

$$\frac{P_t}{F_t} = \left(\frac{D}{d}\right)^2 \frac{P}{F}$$

But from [5]

$$\frac{P_t}{F_t} = E \frac{A}{l}$$

hence

$$\frac{P}{F} = E \frac{A}{l} \left(\frac{d}{D}\right)^2 . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [7]$$

in which the square of the lever index appears as in Equation [1].

9 Proceeding in the same manner,

$$P_t = \frac{D}{d} P$$

in which $P_t = SA$, giving

$$P = SA\left(\frac{d}{D}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [8]$$

and in this expression appears the first power of the lever index as in Equation [2].

10 From [7] and [8]

$$F = \frac{S}{E} l \left(\frac{D}{d}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [9]$$

and evaluating $W = \frac{PF}{2}$ the following expression for work is obtained:

$$W = \frac{1}{2} \frac{S^2}{E} Al \; . \quad [10]$$

which of course is the same whether a lever is used or not used.

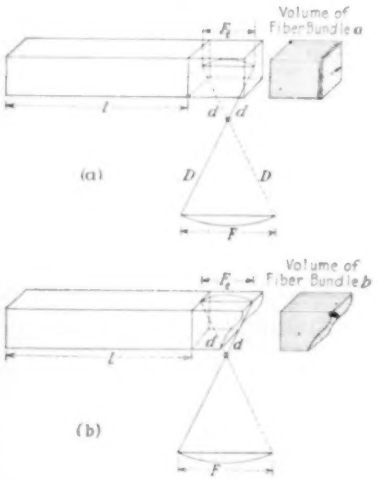

FIG. 3

11 It will be noted that Formulas [7], [8], [9], and [10], derived in a new and different manner, are similar to the generalized orthodox formulas [1], [2], [3], and [4], with the exception that the constants k, m, n and u, respectively, are missing. These constants can be supplied by considering that the minor arm of the lever index is some fractional part of d, which is not identified yet as the depth or diameter of the bar's section. This modification would furnish constants for all formulas except [10], since the resilient energy obtained is independent of the lever system. A constant, less than unity, can be inserted in the latter formula by introducing the idea of stress efficiency, which is assuming that all fibers in the bar are not stretched to the maximum value. As the minor lever arm moves through the distance corresponding to the maximum stretch, even though only a very small portion of the fiber bundle is stretched to this amount, the load P is decreased in proportion. This constant multiplied by 100 might well be called the "per cent efficiency in stressing," and since it

affects the load component of resilient energy, it accordingly affects the load-deflection rate directly. Fig. 3(*a*) shows a 100 per cent efficient fiber bundle and Fig. 3(*b*) shows one of reduced efficiency. The ratio of the volume of *b* to that of *a* multiplied by 100 is the per cent efficiency of stressing in *b*. Although the length of travel is the same for both lever systems, the load is less in *b* because less resistance has to be overcome.

12 Assigning constants, therefore, to Formulas [**7**], [**8**], [**9**], and [**10**] in accordance with the above reasoning and using for these constants the same letters as were used for Formulas [**1**], [**2**], [**3**], and [**4**], namely, *k*, *m*, *n*, and *u*, a set of formulas is obtained that is identical with the latter when viewed from a

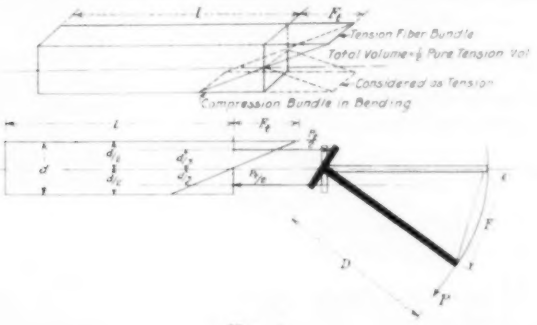

FIG. 4

general standpoint. In their respective order the formulas are:

$$\frac{P}{F} = kE\,\frac{A}{l}\left(\frac{d}{D}\right)^2 \qquad [11]$$

$$P_m = mSA\left(\frac{d}{D}\right) \qquad [12]$$

$$F_m = n\,\frac{S}{E}\,l\left(\frac{D}{d}\right) \qquad [13]$$

$$W_m = \frac{u}{2}\,\frac{S^2}{E}\,Al \qquad [14]$$

13 Thus the same general spring formulas are obtained directly, by what may be considered as the lever principle, as were obtained by generalizing many individual formulas, derived separately by long and somewhat complicated methods. The fact that the arms of the lever describe arcs of circles instead of straight lines does not detract from the merit or logic of the derivation since it is general. It might affect the relative values of the corresponding constants *k*, *m*, *n*, and *u* in both cases, although it must be remembered that in the derivation of nearly every orthodox

spring formula some geometric approximation is made that may be more or less appreciable, depending upon the magnitude of the deflection.

Evaluation of Constants

14 To evaluate the constants k, m, n, and u requires more detailed consideration of the fiber bundle for the various methods of stressing a bar. Take, for example, the type of stressing shown in Fig. 4, in which the pivot of the lever system is placed on the geometric axis of the bar and the minor arm consists of two right-angled extensions. One-half of the stressing is compression, but the correction due to the difference between the tensile and compressive properties of the same material is a refinement hardly necessary at this time, although it would be worthy of a separate consideration. With this qualification, therefore, the value of u might be determined by comparing the volumes shown in Fig. 5. Volume a is equal to one-half of volume b, which indicates that the efficiency in stressing is 50 per cent. To determine k, it must be remembered that this constant involves two factors, one due to a modification of the minor lever arm d and the other due to the reduced efficiency which affects only the load component. If now it is specified that the section of the bar is square, and that the vertical dimension (depth) is d, then the modification factor is $(\frac{1}{3})^2$ since the resultant load on the fiber bundle is concentrated at the center of gravity. The load component is decreased by one-half, as is the efficiency, because the loss in the latter is all taken in the former. Multiplying these two factors would give a value of 1/18 for k. Similar reasoning would indicate a value of $\frac{1}{3} \times \frac{1}{2} = \frac{1}{6}$ for the value of m and 3.00 for the value of n.

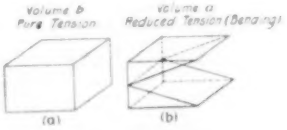

Fig. 5

15 Comparing these values with those of uniform bending, as in helical and spiral springs stressed flexurally, which this type of stressing would seem to approximate, the following results are obtained.

Constant	k	m	n	u
In old derivation	1/12	1/6	2	0.33
In new derivation	1/18	1/6	3	0.50

16 Although these comparisons show appreciable differences, they are not very great, considering that the derivations were made by two absolutely independent methods. The comparisons, however, may have a very important significance, because in more than one instance the exact differences shown above have been found as discrepancies between actual measurements and calculated results of springs with relatively high deflections. Thus A. A. Remington in his paper on The Design and Functioning of Laminated Auto-

mobile Suspension Springs, presented at a meeting of The Institute of Automobile Engineers of England in March, 1922, makes the following statement: "Tests made on spring blades show that the stress cannot be so high as the results given by the unmodified beam formula, as single blades will, without taking a permanent set, carry loads representing a fiber stress as calculated by this formula approximating their tensile strength."

17 The reason Remington gives for the unusual ability of the plate to perform this service is that, "as the relative magnitude of the deflection increases, the fiber-stress section varies from that shown in Fig. 6(*a*) to that shown in Fig. 6(*b*), which is limited in area, depending upon the magnitude of deflection."

18 Applying the author's new method to this idea and using the respective values of the resisting moments shown in Fig. 6, the volume of the bundle in case (*b*) is 50 per cent greater than in case (*a*). Determining the values of constants, k, m, n, and u on this basis results in exactly the same comparative figures as are given in Par. 15.

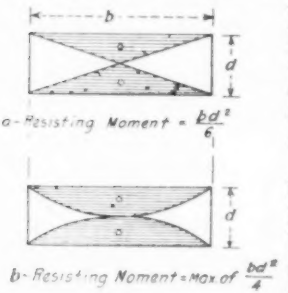

FIG. 6

19 The author does not believe, however, that the fiber section varies in the arbitrary manner pointed out by Remington, although Coker photoelastic tests, if properly carried out, would be convincing one way or the other. Furthermore, the immediate cause given for increase of the stress section is poor stress distribution between different plates in the laminated springs, which does not seem to be a plausible explanation.

20 On the contrary, the author believes that when deflection in such springs is relatively high, the errors due to the geometric approximations made in the derivation of the old formulas become appreciable, but they are on the safe side. If this be true, then it must follow that the constants determined by the new method described in this section of the paper are also in error due to geometric approximations, although this error is of a constant magnitude for all deflections. On the other hand, if the old formulas are in error for any magnitude of deflection, then the indications are that a notable discovery has been made by the author in the theory of flexural springs, because it does not seem possible that two comparisons independently made should give identical results and be merely accidental.

21 Another instance which furnishes confirmation of the author's finding was in the investigation of motor springs for telephone dials. Due to space limitation and operation requirements, these springs, which are of the helical type, worked through

large deflections. Actual test results showed that the regular formulas indicated much too low safe stresses.

22 Finally another instance of confirmation lies in the present practice of using what might be called an empirical modulus of elasticity, which in many cases is taken as 22,500,000 lb. per sq. in. for steel. Granting that a slight modification should be made for the combination of tension and compression that occurs in flexure, the ratio between the value stated and the known value for the modulus of elasticity for steel, namely, 30,000,000 lb. per sq. in., is about 0.75. This ratio checks quite closely with the ratio of the stiffness constants k given in Par. 15.

23 These confirmations of the discrepancies shown in the table seem remarkable enough to warrant a very careful research, both mathematical and experimental, possibly by means of a very sensitive deflection-measuring machine and by Coker tests.

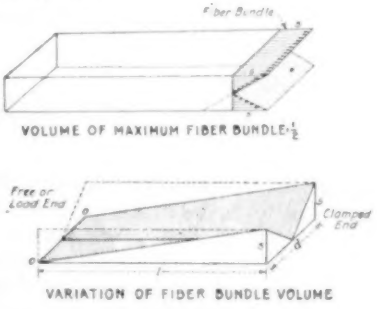

FIG. 7

24 In the case of cantilever flexure, the volume of the fiber bundle is not constant for all sections. Instead, it varies from a maximum at the clamped end to zero at the free or loaded end as shown in Fig. 7. The volume of the total fiber bundle can be calculated very easily by means of this sketch, it being equal to one-third of the total volume for pure tension. Since the average value of the maximum stresses in the sections is $(S+0)/2$, the value of u is determined as follows:

$$W_m = \frac{u}{2}\,\frac{S^2}{E}\,Al = \frac{1}{3} \times \frac{(S/2)^2}{2E}\,Al$$

or

$$W_m = \frac{1}{3} \times \frac{1}{4} \times \frac{S^2}{2E}\,Al = \frac{1}{12}\,\frac{S^2}{2E}\,Al = 0.083\,\frac{S^2}{2E}\,Al$$

25 The value of m is $\frac{1}{2} \times \frac{1}{3} = \frac{1}{6}$, the factor $\frac{1}{2}$ being due to the average stress in the maximum section and the factor $\frac{1}{3}$ being the minor lever-arm modifier. From u and m the values of k and n are found to be 0.35 and 0.48, respectively. The comparison of these values is shown below.

Constant	k	m	n	u
In old derivation	0.25	0.16	0.64	0.10
In new derivation	0.35	0.16	0.48	0.08

26 Turning now to that type of stressing which produces a fiber bundle in a bar of circular cross-section as shown in Fig. 8, the constants were determined as in the previous manner, except with the additional modification of changing the load from tension

to shear, the comparative results shown below being thus obtained. The value of k is the product of $(\frac{1}{3})^2$, the modifier for the minor lever arm, $\frac{2}{3}$, the fiber energy modifier, and $\frac{2}{3}$, which converts the tensile load to a shear load. Similarly m is the product $\frac{1}{3} \times \frac{2}{3} \times \frac{2}{3}$, the last fraction being the modifier for converting tensile to shear stress. The constant n is equal to m/k.

Constant	k	m	n	u
In old derivation	0.05	0.16	3.20	0.50
In new derivation	0.03	0.14	4.73	0.66

27 Thus with the above figures and the results given in Pars. 15 and 25, it can at least be claimed that the new method of derivation for the generalized spring formulas is sound and logical. Further, it can be said that since in the case of flexural springs the discrepancies shown in Par. 15 have been confirmed by actual tests, the entire question of whether the present spring formulas are sufficiently accurate in all cases is in need of considerable research. This finding might have been suspected, due to the fact that many spring formulas are taken from works on mechanics of materials in which structural members of high rigidity are considered. Until such research has either confirmed or disproved the new values of the constants, the values derived from orthodox formulas will be used in the tentative code to follow. It is to be remembered, however, that the new derivation of the generalized formulas is fully applicable in any case up to the point of valuation of constants for specific cases.

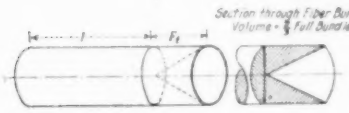

FIG. 8

28 Fig. 9 illustrates the relative values of the constants k, m, n, and u for several methods of stressing. The basis of comparison is that the bar dimensions, maximum stress, and lever index are the same in all cases. Thus in the tensile triangle of resilient energy the load and deflection legs are both equal to unity, while in the torsional triangle the deflection leg is greatly increased at the cost of 50 per cent efficiency in stressing. In the energy triangle for cantilever flexure the efficiency is only 8 per cent, and nothing is gained for the sacrifice of 92 per cent efficiency since the deflection and load legs are both less than unity. Uniform flexure is therefore preferable to the latter type of flexure when considered on this basis.

29 It is unfortunate that the stress efficiency is unavoidably decreased by the various methods of stressing, because a sufficiently large gain in flexibility is not obtained thereby. The application of the lever principle to the tension bar shows that the flexibility is greatly increased without a loss in stress efficiency. Consider, for example, the case of uniform flexure, whose constants are given in Fig. 9. At a loss of 75 per cent in stress efficiency a gain of

only 100 per cent in flexibility is obtained, although it is true that the load-deflection rate is reduced to $\frac{1}{16}$ of that for pure tension. Now by the application of a lever index equal to 20 — not an unusual value for actual springs — no loss in stress efficiency is incurred, the flexibility is increased twenty-fold, and the load-deflection rate is decreased to 1/400 of that for pure tension.

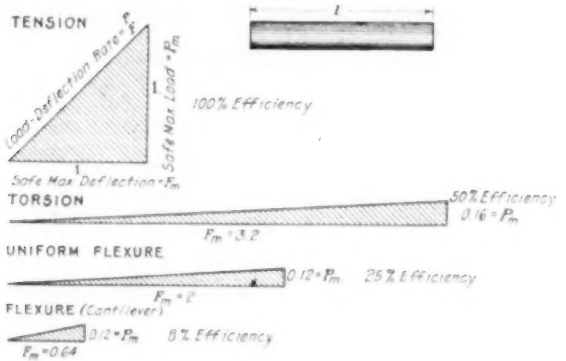

FIG. 9

30 With the foregoing principles established on a logical and sound basis, the author feels that the following tentative code, which is constructed in accordance with these principles, should prove useful and valuable. Until research has either confirmed or disproved the new values of the constants the values derived from the orthodox formulas will be used in the code. It should be remembered, however, that the new derivation of the generalized formulas is fully applicable in any case up to the point of valuation of constants for specific cases.

TENTATIVE CODE OF DESIGN FOR MECHANICAL SPRINGS

GENERAL DEFINITIONS

31 A mechanical spring is an elastic body whose load-deflection rate and maximum safe deflection are of values suitable for mechanical use.[1]

32 A mechanical-spring material is an elastic material of the kind which, when made into bodies of a shape and size suitable for use in mechanical design, will function repeatedly and permanently as a mechanical spring.

[1] This definition is an abridged form of the one given in the author's paper on mechanical springs in which "maximum safe deflection" replaces "elastic range of deflection" so as to conform with the terms used in the formulas.

Mechanical Service Requirements

33 A certain point in a mechanical member having practically a rigid construction shall require contact with the load or free end of a spring in such a manner as to exert a pull or pressure of P_0 lb. After a travel of F_1 in., a pull or pressure of P_1 lb. shall be exerted. The load-deflection rate required of the spring shall therefore be equal to $\frac{P_1-P_0}{F_1}$ as illustrated in Fig. 10.

34 The motion of the point of contact shall be either transla-

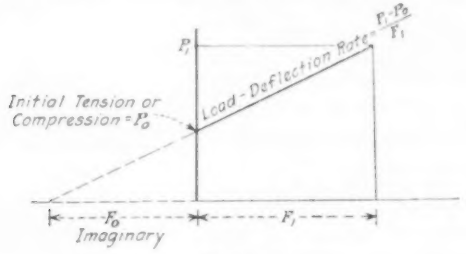

Fig. 10

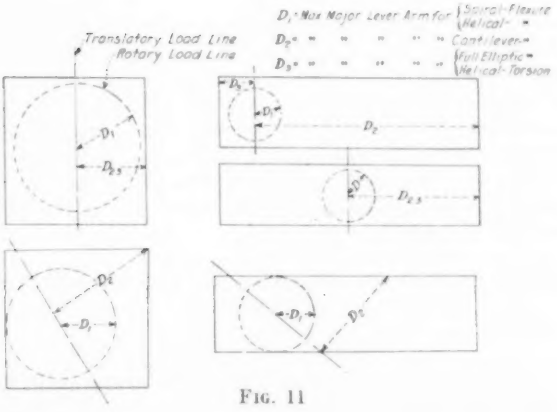

Fig. 11

tory or rotatory and shall fall within a satisfactory range of periodicity. Types of springs giving translatory motion are helical, stressed torsionally, and flat, stressed flexurally, while springs giving rotary motion are those of the helical and spiral types stressed flexurally.

35 The spring selected shall perform the required service within the space assigned. The space-limitation rectangle and the load line or path of motion of contact point are fixed for specific service requirements. Various cases which might arise are shown in Fig. 11.

36 Service requirements involving the suitability of the spring material other than from the spring-design standpoint, such as corrosion, magnetic property, etc., are considered elsewhere under the heading of Materials, Par. 46.

FORMULAS [1]

Required Constant Material Bar Lever ratio

Load-deflection rate:

$$\frac{P}{F} = k \times E \times \frac{A}{l} \times \left(\frac{d}{D}\right)^2 \qquad [\mathbf{I}]$$

Safe maximum load:

$$P_m = m \times S \times A \times \left(\frac{d}{D}\right) \qquad [\mathbf{II}]$$

Safe maximum deflection:

$$F_m = n \times \frac{S}{E} \times l \times \left(\frac{D}{d}\right) \qquad [\mathbf{III}]$$

Safe maximum work:

$$W_m = \frac{u}{2} \times \frac{S^2}{E} \times Al \qquad [\mathbf{IV}]$$

Other formulas are —

Total weight of spring material:

$$w = (Al \times \delta) + \text{elastically dead material} \qquad [\mathbf{V}]$$

Weight per linear foot:

$$\frac{w}{l} = (A \times \delta) + \text{elastically dead material} \qquad [\mathbf{VI}]$$

Period of oscillation: $T = \frac{2\pi}{\sqrt{g}} \sqrt{\left(K^2Q + \frac{w}{i}\right)\frac{F}{P}}$ [VII]

37 The above formulas can be used to calculate any type of mechanical spring to meet a given service requirement when the terms are fully understood and the values of the constants are known. These terms are to be used as follows:

SERVICE REQUIREMENTS

38 The diagram in Fig. 12 shows the significance of the load-deflection rate P/F, safe maximum load P_m, safe maximum deflection F_m, and the safe maximum work W_m. The units used are pounds and inches, an inch-pound being the measure of work or resilient energy.

39 The total weight of spring material w plus elastically dead material is in pounds. By "elastically dead material" is meant that material which does not furnish elastic energy, such as the

[1] Formulas [I]-[IV] are the same as Formulas [1]-[4] or [11]-[14].

clamped portions of automotive leaf springs, the squared or looped ends of helical springs, and also the slight wastage involved in manufacture. "Total weight per linear foot" means the weight of a section of bar one foot long, assuming the cross-section to be constant. The latter figure is useful in computing the total cost of the spring bar or plate required.

40 The period T is the number of seconds required for one complete oscillation of the load end of the spring while either

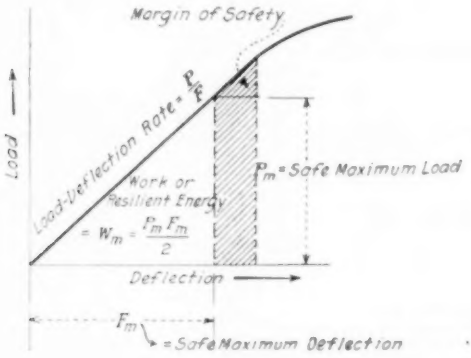

FIG. 12

constantly carrying a weight of Q lb. or just its own weight w lb., i.e., vibrating freely.[1]

LEVER RATIO

41 The lever ratio or index D/d consists of the minor arm d and major arm D, both in inches. For simplicity, the minor arm is made equal to the depth or diameter of the bar or plate section, but actually this arm varies directly from zero to $d/2$, the resultant value being $d/3$. The major arm D is in general equal to the distance from the load axis, whether it be a straight line or a circle, to the clamped end or geometrical axis of the spring. Fig. 13 gives four of the principal types of springs showing the major arm D. In the case of the hour-glass, barrel, and conical types of helical springs, whether stressed torsionally or flexurally, the mean radius of the maximum-sized coil is taken as D and a factor is included in each of the constants k, m, and n allowing for variation in the sizes of the other coils.

[1] This general periodicity law for mechanical springs was derived by the author in 1922 and published in the *American Machinist*.

Bar

42 The cross-sectional area of the bar, equal to A sq. in., may have any of the shapes shown in Fig. 14, the dimensions being as shown. In the case of rectangular and elliptical sections stressed in torsion, a factor J equal to the ratio b/d is included in the constants k, m, n, and u. Likewise, in the case of flat laminated springs that have trapezoidal plan developments, a factor s, equal

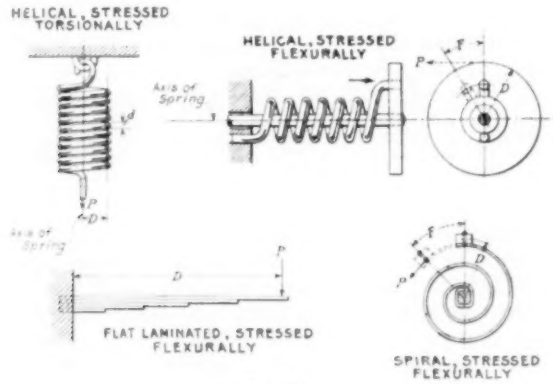

Fig. 13

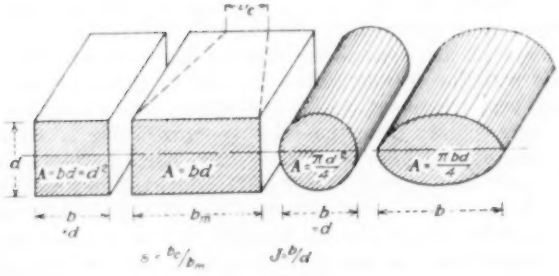

Fig. 14

to the ratio b_c/b_m, is included in the same constants. The minimum width at the free-end section is b_c and the width of the maximum section at the clamped end is b_m. The total length of the bar or plate, excluding the elastically dead portion, is l in. In flat laminated springs this length l is obviously equal to the major arm D of the lever ratio, while in other types of springs, such as the helically or spirally wound types, the relation of l to the mean radius of the coil R and the number of coils or convolutions is determined by ordinary geometry.

43 For the helix,

$$l = 2\pi RN$$

where N equals the number of coils. In the case of the torsionally stressed type, l is equal to $2\pi DN$.

44 For the spiral,

$$l = 2\pi \frac{(R+r)}{2} N \text{ (approximately)}$$

where R is the radius of the outside convolution and r that of the inside convolution, while N equals the number of convolutions. The greater the number of convolutions, the greater will be the accuracy of the above approximation.

45 Al is the volume of the active material in the bar in cubic inches, while $\frac{A}{l}$ is the bar ratio.

Material

46 The tensile or Young's modulus of elasticity in lb. per sq. in. is designated by E. The torsional modulus of elasticity G in lb. per sq. in. is usually equal to about two-fifths of E, and the values of the constants k, n, and u in Table 1 are based upon this factor. Values of E given in Table 2 for different metals show that they range from 6,000,000 to 60,000,000 lb. per sq. in. and that they tend to vary in accordance with the corresponding densities δ. The latter figure is used in connection with the calculation of weights of spring material required, as pointed out in Par. 39. Very few conditions appreciably affect the value of Young's modulus for a given material, and even these to only a slight extent. Steel, with its wide variation in microstructure, due to variations in carbon content, heat treatment, and cold working, has a modulus which differs only slightly from the value given in Table 2. Of course, in the case of a high-nickel-chrome steel, which has only about a 60 per cent iron content, a noticeably different modulus will be found. Similarly in the non-ferrous alloys an appreciable variation in the modulus would also be expected, due to a wider variation in the composition. In general, so far as composition, and possibly cold working, are concerned, it may be said that if the variation in the constituents or compactness is sufficient to change the density appreciably, then the modulus will also be changed appreciably.

47 The slight deviation from proportionality between stress and strain produced in a specimen by overstrain may cause slight inaccuracies in the plotting of the stress-strain curve and consequently in the modulus of elasticity. Low-temperature annealing would cure this imperfect elasticity, allowing a more accurate determination of the modulus which, when compared with the value previously determined, would show a slight difference. Moduli determined for specimens having the direction of rolling oriented

TABLE 1 STRESS-METHOD AND FORM-OF-SECTION CONSTANTS

Type of Spring	Method of Stressing	Form of Section	k	m	n	u	Per Cent Efficiency in Stressing
Helical	Torsion	Round	0.05	0.16	3.20	0.50	50
"	"	Square	0.08	0.19	2.25	0.42	42
"	"	Elliptical	$0.10\left(\frac{J^2}{1+J^2}\right)$	$0.22\left(\frac{J^2}{\sqrt{1+J^2}}\right)$	$2.3\left(\frac{\sqrt{1+J^2}}{J}\right)$	$0.50J$	$50J$
"	"	Rectangular	$0.16\left(\frac{J^2}{1+J^2}\right)$	$0.27\left(\frac{J^2}{\sqrt{1+J^2}}\right)$	$1.6\left(\frac{\sqrt{1+J^2}}{J}\right)$	$0.42J$	$42J$
Conical	"	Round	0.10	0.16	1.60	0.25	25
"	"	Square	0.16	0.19	1.12	0.21	21
Helical	Flexure	Round or Elliptical	0.06	0.12	2.0	0.25	25
"	"	Square or Rectangular	0.08	0.16	2.0	0.33	33
Spiral	"	Round or Elliptical	0.06	0.12	2.0	0.25	25
"	"	Square or Rectangular	0.08	0.16	2.0	0.33	33
Cantilever	"	Round or Elliptical	0.19	0.12	0.65	0.08	8
"	"	Square or Rectangular	0.25	0.16	0.65	0.10	10
Cantilever, Triangular Plan[1]	"	Rectangular	0.16	0.16	1.00	0.16	16
Cantilever, Trapezoidal Plan[1]	"	"	$1.00\left(\frac{s+2}{12}\right)$	0.16	$\frac{2}{s+2}$	$\frac{0.32}{s+2}$	$\frac{32}{s+2}$

Note: $s = \frac{b_e}{b_m}$ (see Fig. 14)

$J = \frac{b}{d}$ (see Fig. 14)

Assumed: $G = 0.4E$ and $S = 0.63S$ for tension.

[1] Basic for laminated springs.

TABLE 2 VALUES OF MATERIAL INDEX AND OTHER PROPERTIES FOR VARIOUS METALS

Reference Number	Material	Form of Material	Density in Lb. per Cu. In. = δ	Young's Modulus in Lb. per Sq. In. = E	Proportional Limit in Lb. per Sq. In.	Material Index (Static)	Endurance Limit in Lb. per Sq. In.	Material Index (Kinetic)	Tensile Strength in Lb. per Sq. In.	Ult. Elongation, Per Cent in 2 in.	Percentage Reduction in Area
1	Magnesium alloy (Zn—4%)	Bar	0.065	6,000,000	6,000	0.0010	.....	.....	36,000	17	20
2	Duralumin	Bar	0.100	10,000,000	25,000	0.0025	.....	.....	50,000	16	50
3	High-nickel-chrome steel	Rolled sheet	0.218	27,000,000	100,000	0.0037	.....	.....	127,000	50	55
4	High-nickel-chrome steel	Cold-drawn wire	0.220	28,000,000	180,000	0.0064	.....	.....	220,000	12	..
5	Chromium steel	Rolled sheet	0.278	29,000,000	150,000	0.0052	.....	.....	185,000	15	35
6	Low-nickel-chrome steel	Bar	0.280	30,000,000	115,000	0.0038	.....	.....	138,000	18	62
7	Spring carbon steel (1.0% C.)	Rolled sheet	0.280	30,000,000	152,000	0.0056	.....	.....	168,000	11	..
8	Carbon steel (1.20% C.)	Bar	0.280	30,000,000	120,000	0.0040	92,000	0.0036	180,000	9	15
9	Steel music wire	Drawn wire	0.285	30,800,000	260,000	0.0084	.....	.....	310,000	8	..
10	Silicon-manganese steel	Bar	0.282	30,000,000	100,000	0.0033	65,000	0.0022	157,000	16	40
11	Spring molybdenum steel	Sheet	....	30,000,000	215,000	0.0071	.....	.....	225,000	14	40
12	Nickel steel	Bar	0.280	30,000,000	82,000	0.0027	63,000	0.0021	112,000	24	60
13	Pure tungsten	Small drawn wire	0.720	60,000,000	370,000	0.0051	.....	.....	490,000	..	..
14	Brass (Zn—35%)	Rolled sheet	0.307	12,000,000	40,000	0.0033	.....	.....	60,000	25	30
15	Phosphor bronze (8% Tin)	Rolled sheet	0.312	16,000,000	70,000	0.0044	40,000	0.0025	90,000	9	..
16	Nickel silver (Ni—18%, Zn—12%)	Rolled sheet	0.320	17,000,000	65,000	0.0038	.....	.....	90,000	3	33
17	Monel metal	Rolled sheet	0.320	23,000,000	50,000	0.0022	.....	.....	94,000	30	35

TABLE 3 VALUES OF i FOR DIFFERENT FORMS OF SPRINGS

Directions of Oscillations	Form of Spring	Condition of Ends F = Fixed R = Free	Value of v for the Fundamental Mode	Values of v for the Higher Modes	Values of i for Any of the Modal Values m	Values of i with Light or without Any Attached Weight v = Fundamental
Longitudinal	Helical	$1F$, $1R$	1	3, 5, 7, 9, etc.	$4 \div \pi^2 v^2$	0.40
	Helical	$2F$ or $2R$	2	4, 6, 8, 10, etc.	$4 \div \pi^2 v^2$	0.40
Transverse	Cantilever	$1F$, $1R$	1.194	3, 5, 7, 9, etc.	$48 \div \pi^4 v^4$	0.25
	Buffer	2 supported	2	4, 6, 8, 10, etc.	$768 \div \pi^4 v^4$	0.50
	Buffer	2 free				
		Consider one overhanging end	1.194	3, 5, 7, 9, etc.	$48 \div \pi^4 v^4$	0.25
	Spiral	$1F$, $1R$	1.194	3, 5, 7, 9, etc.	$48 \div \pi^4 v^4$	0.25
	Coiled	$1F$, $1R$	1.194	3, 5, 7, 9, etc.	$48 \div \pi^4 v^4$	0.25

differently may also show differences, as will be readily understood.

48 Probably the greatest variation in the modulus for a metal of a given composition is produced by variations in the surrounding temperature. Thus, the modulus for steel is reduced about 10 per cent by elevating the surrounding temperature from about 70 deg. fahr. to 300 deg. fahr. This fact should be taken into consideration, especially when designing springs for steam and gasoline apparatus such as turbines and reciprocating engines.

49 The maximum safe working stress used in machine-design calculations is usually determined by dividing the ultimate strength of the material under consideration by a factor of safety. This factor is estimated by judgment based upon the possible maximum service conditions to be expected, upon the unknown properties of the materials, such as resistance to fatigue and shock failure, and finally upon the commercial variations of the known properties. To increase the margin of safety, it is often the practice to use a material of considerable ductility. However, this property is usually accompanied by low strength, which accounts for the rapid growth of the use of alloy steels having both high strength and high ductility.

50 The proportional limit not being considered in ordinary machine-design calculations of the maximum safe stress, permanent set may or may not take place. If it does, the small displacement of the material is usually not detrimental to the proper functioning of the machine, but on the other hand there are some types of mechanisms in machine design, and this is particularly true of springs, in which displacement of material due to permanent set is often detrimental. For this reason the maximum safe working stress is usually based upon the proportional limit S in spring design, and for the present the safe stress will be considered equal to this limit, after which the manner of allowing for the unknown factors such as fatigue, impact, etc., will be explained.

51 The value of S varies over a very large range due to many conditions, such as slight variations in composition, heat treatment, cold working, surrounding temperatures, etc. Greater care should therefore be exercised in ascertaining the correct value of S than in finding the value for E because, as pointed out previously in consideration of the load-deflection rate, the value of the modulus E varies but slightly under these different conditions. The value of S in torsion is generally equal to about 0.63 of the value of S in flexure or tension, and this factor is included in constants m, n, and u for springs stressed torsionally. In any case, for an accurate comparison of different metals the values of S and E should not be used unless they actually represent the metals under consideration. The table, which gives these values for tensile stressing along with the corresponding material index and

other properties, should be used only for general purposes, as the figures were collected from many diverse sources.

52 Up to this point the working stress S has been assumed equal to the proportional limit, which does not provide any margin of safety against permanent deformation and fatigue failure. The latter type of failure in the case of stress reversals very often takes place below the proportional limit, hence the manner of allowing for this important phenomena will be discussed first. In the strict theoretical sense, the assumption that the proportional limit is the value to substitute for S in regard to fatigue is probably correct, as recent fatigue investigations indicate that the endurance limit, i.e., the maximum stress which will not cause progressive failure, corresponds, to a " true " proportional limit which can only be determined statically by very refined methods. However, Gough [1] of the National Physical Laboratory in seeking to shorten the long-drawn-out fatigue tests found just such a " true " proportional limit by plotting a stress-strain curve in which the loads and deflections were measured while the specimen under test was going through rapid reversals of stress. Likewise Stromeyer [2] found this limit by plotting gradually increasing stresses against the corresponding temperatures resulting from the heat generated in the specimen while it was being put through rapid reversals of these stresses. Both of these methods gave stress values which agreed very closely with the endurance limits determined on the same materials by prolonged fatigue tests. Although the aim in ideal spring design is to have pure elastic action throughout the maximum safe range of deflection F_m, this cannot be fully assured until the present fatigue investigations going on in various countries, principally in the United States and England, have determined the endurance limits for most spring metals. However, where a spring is not deflected more than, say, one million times, it is not really essential in most cases to have a pure elastic range, inasmuch as the ordinary proportional limit determined with the present-day type of testing machine is sufficiently accurate to limit the safe deflection range. In fact, if the " true " limit were to be exceeded occasionally in service, i.e., if the spring were only occasionally overstrained, an apparently new and slightly higher limit would be established which would increase the allowable deflection range.

53 In view of the above considerations and the fact that the " true " proportional or endurance limits of those metals, for which the fatigue investigations have shown such limits to exist, do not bear any definite relation to the statically determined proportional limits, the author has suggested that the material index $\frac{S}{E}$

[1] H. J. Gough, B. Sc., Improvements in Methods of Fatigue Testing. *The Engineer*, August 12, 1921.

[2] C. E. Stromeyer, Memorandum of the Chief Engineer of Manchester Steam Users Association, 1913.

be divided into three types: the *static* index, to be used in spring design where the number of deflections is less than one million; the *kinetic* index, to be used where the number of deflections is considerably greater than one million; and the *elevated-temperature* index, to be used at temperatures above 500 deg. fahr. Table 2 shows the value of the static material index for each of the metals listed and of the kinetic material index for those metals whose endurance limits have been determined. The modulus is the same for both of these indices, because fatigue apparently does not affect this property.

54 Both the modulus and the proportional limit are decreased appreciably when the surrounding temperature is increased, but the proportional limit usually decreases to a much greater extent, giving rise to a decrease in the elevated-temperature index. The character of this variation would be different for different metals, which necessitates a special determination of both S and E at the temperature desired, in order to compute the material index. Indices for both monel metal and high nickel-chrome steels at 1000 deg. fahr. are probably not less than 60 per cent of the indices at room temperature, as compared with about 40 per cent for some ordinary carbon steels. Springs required to operate at high temperatures should be made of metals whose material indices will not decrease too much.

55 In actual spring design there are additional factors which determine the maximum safe working stress, such as occasional overload, sudden loads, and impact loads, but which need not be considered in the comparison of two metals unless the material indices are somewhere nearly equal in value. Then the metal having the greatest ductility as gaged by the percentage elongation or by such empirical tests as the Erichsen and simple bending tests for flat spring materials, or by the swaging and twisting tests for helical spring material. This judgment should be further influenced by the ratio between the ultimate tensile strength and the proportional limit, and also by the percentage reduction in area and the impact-test figure. In any case the ductility should be sufficient because plastic action distributes the stress uniformly, while sufficient strength will prevent breakage when local overstressing or impact loads take place.

56 In accordance with the above principles the static, kinetic, or elevated-temperature material indices, as the case may be, should be modified by a factor of safety giving what might be called the "safe working material index" equal to S/E for use in Formula [**III**]. S and S^2/E should be treated similarly before using in connection with Formulas [**II**] and [**IV**], respectively.

57 S^2/E is the modulus of resilience in inch-pounds per cubic inch. Since S is squared in this term, and since it is subject to greater variation than E, the variation of this modulus will be much greater than in the case of the material index.

58 Service requirements sometimes specify freedom from corrosion due to various influences. Atmospheric corrosion would prohibit in certain cases the use of the best kind of spring metal, namely, steel. For small steel springs it is possible, although difficult, to protect against such corrosion by plating the steel with tin, nickel, or a combination of nickel-copper-nickel, while for medium-sized springs protection for a short time may be had by painting. This problem is solved more satisfactorily in cases where space limitations permit by resorting to the use of non-rusting steels, phosphor bronze, nickel silver, monel metal, nickel, and brass. The last-named metal is subject to season cracking, however, particularly if under sustained stress.

59 Steel as a spring material is prohibited also where a spring must be non-magnetic, except in the case of normally austenitic steel, which, however, has an inferior material index and modulus of resilience.

Constants k, m, n, u

60 These constants vary with method and efficiency of stressing, form of bar or plate cross-section, and with non-uniformity in size of bar or plate cross-section. The values of these constants for some of the most important types of springs are given in Table 1. As stated in Par. 42, these constants may contain either a factor s or J, or both. Factor s is the ratio of the minimum width b_c of bar or plate having a trapezoidal form and of constant thickness d to the maximum width b_m. The area A used in the formulas in this case is $b_m d$. Factor J is the ratio of the breadth b to the depth d of rectangular and elliptical sections of bars stressed torsionally.

Special Terms and Constants

61 Q is the constant weight in pounds that an oscillating spring carries, such as the sprung weight of an automobile or locomotive. If the spring vibrates freely Q equals zero and hence vanishes along with K^2. Where Q is large compared with w or the weight of the spring, w/i practically vanishes.

62 The gyration factor K is equal to unity in all cases except that of the balance wheel on either spiral or coiled springs, that is, where the weight Q gyrates about the axis of the spring. In these cases K is equal to the radius of gyration divided by D. This is a particular case of a circular load line referred to in a paragraph on Service Requirements.

63 The term i is called the "equivalent mass factor," and its value depends upon the mode factor V. Values of i for the various types of springs vibrating freely in the fundamental and higher modes are given in Table 3.

64 The constant g is the acceleration of gravity, which is taken as 386 in. per sec. per sec.

BRIEF SUMMARY OF TERMS AND CONSTANTS IN FORMULAS

$\frac{P}{F}$ = load-deflection rate, lb. per in.
P_m = load, safe maximum, lb.
F_m = deflection, safe maximum, in.
W_m = work, safe maximum, in-lb.
$\frac{S^2}{E}$ = modulus of resilience, in-lb. per cu. in.
$\frac{S}{E}$ = material index, safe working (pure number)
S = unit tensile stress, safe maximum, lb. per sq. in. (= 1.6S for torsion)
E = tensile modulus of elasticity, lb. per sq. in. (= 2.5G, where G = modulus for torsion)
Al = volume of bar, cu. in.
$\frac{A}{l}$ = bar ratio, in.
A = area of bar cross-section, sq. in.
l = length of bar, in.
$\left(\frac{D}{d}\right)$ = lever index, (pure number)
D = major lever arm or moment arm of load, in.
d = minor lever arm (twice maximum) or depth of section, in.
k = load-deflection-rate constant
m = safe-maximum-load constant
n = safe-maximum-deflection constant
u = safe-maximum-work constant
w = total weight of spring material, including elastically dead material, lb.
δ = density of spring material, lb. per cu. in.
T = period of oscillation in seconds
g = acceleration of gravity = 386 in. per sec. per sec.
Q = weight of mass attached to oscillating spring, lb.
K = gyration factor, equal to the radius of gyration divided by D (pure number)
i = equivalent mass factor depending on mode of oscillation (pure number).

GENERAL PROCEDURE IN DESIGN OF SPRING

65 Referring to Fig. 11,

1 Determine maximum area and form of space allowed for spring in machine or mechanism.

2 Determine location, direction, and shape of load path required.

3 From operation requirements of machine or mechanism determine load-deflection rate, maximum travel, or load, and if important, the period of oscillation. (See Par. 38)

4 Select types of springs that fulfill 1 and 2 and then for each type calculate the spring dimensions, using Formulas [**I**] and [**III**] (or [**II**]), to fulfill 3. Using Formula [**IV**], check the results and select the best spring of the various types calculated.

5 The following general precautions should be observed.

a The material constants E and S for the proposed spring material should be known as accurately as possible

b The lever index D/d for helical springs should not be less than 1.5 in any case, and preferably not less than 3. Between 1.5 and 3 the value of the lever index is permissible only when unavoidable, because of space limitations and other causes

c The manner of securing or clamping the inactive part of springs should be considered very carefully in order to eliminate high local stresses. This is likely to occur in the end loops or squared ends of helical springs and at the bands of elliptic springs

d The unequal distribution of stress in laminated springs, due to deviation of the actual developed plan view from the theoretical plan view and to the practice of "nipping" during the process of manufacture should be given very careful attention. The same precaution should likewise be taken in the case of high localized stresses due to nipping, clipping, and the practice of putting center pins through spring blades.

Conclusion

66 In the foregoing exposition of the author's general method of spring design in brief codified form, preceded by a new and simple derivation of the general formulas that emphasizes the logic and simplicity of the method, it is felt that an initial start has been made toward the formulation of a permanent code of design for mechanical springs. No attempt has therefore been made to present a code complete in every detail.

67 In addition to establishing the general method of design on a sound logical basis, the new and direct derivation of the generalized spring formulas has given, in the specific case of flexural springs, results that agree with those obtained from actual tests made by various investigators, but which disagree with those calculated from the regular formulas.

DISCUSSION

T. McLEAN JASPER.[1] The idea of stiffness and the use of the load-deflection rate are factors most necessary in spring design and these have been introduced into the author's formulas in a most usable manner. In Formula [4] the expansion could very well be carried out to include the generalized case, in which instead of $\frac{AlS^2u}{2E}$ the generalized energy formula for elastic bodies could be introduced.[2] The generalized formula would then become

$$W_m = \frac{u}{2E} \int \{[S_x^2+S_y^2+S_z^2]-2\rho[S_xS_y+S_yS_z+S_zS_x]+ 2(1+\rho)[S_{xy}^2+S_{yz}^2+S_{zx}^2]\}dv$$

where S_x, S_y, and S_z = normal unit stresses in the x, y and z directions

S_{xy}, S_{yz} and S_{zx} = corresponding torsional stresses

E = modulus of elasticity in tension or compression

ρ = Poisson's ratio for the material

dv = the volume element.

This formula would be correct for all kinds of springs and is equivalent to saying that the safe maximum work that could be performed by a spring is equivalent to the total energy that can be absorbed by it with safety. This generalized formula should be readily applicable to any shape of spring and would be much modified when applied to springs of any one type. The constant u for any material, the writer believes, can be found only by dynamical experiments for any type of spring.

It seems also that W_m for springs should be controlled by the endurance limits and not by the elastic limits of the materials. For springs it is undoubtedly necessary to use as much of the elastic range of the material as possible in order that they may perform their work most efficiently. The use of the energy relations in the testing of materials has been outlined by the writer.[3] The converse of this, the use of tests on materials for the application of the energy relations, is just as logical. Such a course eventually would lead to a few simple tests on materials proposed for use in springs, and these tests immediately would determine the suitability of such materials for springs and the amount of work per unit that the materials safely could perform.

[1] Engineer of Tests, Fatigue of Metals Investigation, University of Illinois, Urbana, Ill.

[2] H. M. Westergaard, Buckling of Elastic Structures, A.S.C.E. Proceedings, vol. 85, 1922; A. E. H. Love, Mathematical Theory of Elasticity, 1920 Edition.

[3] *Philosophical Magazine*, October, 1923.

In this manner, referring again to Formula [4], W_m would be a constant obtained from tests for any one material and heat treatment. The application of the generalized energy relation would depend on the shape and design of the spring. u would be an experimental constant for a particular shape of spring, and A and l (or dv) would be factors to be selected by the designer to give the deflections needed in the structure to which the spring is to be applied. The question of the effect of temperature on the moduli of elasticity and rigidity or torsional modulus and Poisson's ratio has also been taken up by the writer in the *Philosophical Magazine* referred to above.[1]

The writer believes that this paper is a long step forward in developing a logical basis for spring design.

W. G. BROMBACHER.[2] The author's idea that the separate formulas for the various types of springs, such as helical, conical, cantilever, etc., can be grouped into one set of formulas which apply to all classes of springs is new and valuable. Each of the general formulas given in Equations [1] to [4] contains a factor which varies in value for each type of spring. This new development should prove to be a powerful tool, both for the proper selection and design of springs and in aiding the attack on the general spring problem.

The author shows that the values of the constants for the various types of springs differ from those given by orthodox spring formulas, but modification of the value of the constants will not affect the usefulness of the general formulas. The constant k in the load-deflection formula, Equation [1], for helical springs for tension and compression has, as derived by the author, the value of 0.03 instead of the orthodox value of 0.05. This means that the stiffness (load-deflection rate) as given by the new value of k is 60 per cent of that given by the old value. The writer can show that for a particular group of phosphor-bronze helical springs in compression the stiffness is from 5 to 20 per cent less than that given by the formula in which k equalled 0.05. The use of the value 0.03 for k gives a discrepancy between theory and experiment of from 20 to 35 per cent in the other direction. Hence, 0.03 seems to be too low a value.

Attention also should be called to the fact that the value of the stress method and form-of-section constants given in Table 1 will be subject to modification in cases where the stress is torsional. This is due to the fact that the modulus in torsion has been assumed as $\frac{2}{5}$ of the value of Young's modulus, E. Deviations from this assumption exist which are of practical importance. Certain factors of performance of springs which are not mentioned

[1] See also Proceedings, A.S.T.M., vol. 24, Part II, 1924.

[2] Associate Physicist, Bureau of Standards, Washington, D. C.

in the code of design, probably because of lack of data, are of particular importance in the field of precision-instrument design and perhaps in other fields. In connection with these factors no formulas or comparative information on various materials are available to the designer. Nevertheless, these are factors worthy of mention in the code of design. They are hysteresis, after-effect, and change of stiffness (load-deflection rate) with temperature.

Hysteresis is the difference in the deflection of a spring for a given load in a load-deflection cycle in which the load is first increased to a definite value at a given rate, and is then decreased to its original value at the same rate. It is desirable that the hysteresis of a spring be a minimum in all cases where the reading of an instrument depends on the deflection of a spring.

After-effect is the difference in the deflection of a spring at the beginning and at the end of a load cycle. In instruments, after-effect would be characterized by the failure to return to zero position after the removal of an applied load from the operating elastic system.

Change of stiffness (load-deflection rate) with temperature is undoubtedly caused by a change in the modulus E or G. It is certain that this does not cause the entire change. Relatively small percentage changes are of importance since the change in stiffness affects directly the scale value[1] of the instrument.

This temperature effect requires that a spring material be used that gives a minimum change in load-deflection rate when temperature changes are anticipated, as in aeronautic instruments, for instance.

As both hysteresis and after-effect are greatly influenced by the rate of loading and the length of application of the load, time must enter into their specifications. Both can be reduced for a given material and a given rate of loading by reducing the allowable maximum fiber stresses.

DAYTON A. GURNEY.[2] The Ordnance Department as a consumer of springs is much interested in the development of a standard method of design. The requirements for springs used by the Ordnance Department, particularly for counter recoil springs and equilibrator springs, are quite severe, and it is important that the loads and deflections be held within narrow limits. It has been the experience of the Ordnance Department that there is a marked difference in the torsional modulus for springs made of various kinds of steel. For instance, some springs have been obtained in which this torsional modulus was as low as 6,000,000

[1] Scale value of an instrument is defined as the ratio of the true pressure change to the indicated pressure change.

[2] Ordnance Engineer, Ordnance Office, U. S. A., Washington, D. C. Mem. A.S.M.E.

instead of 10,000,000 or 11,000,000 as ordinarily used. The torsional modulus of a spring when the load is being released has been found to be about 11 per cent less than when the load is being increased. This so-called hysteresis effect is quite detrimental in the case of equilibrator springs, which are used to balance the elevating parts of a gun carriage, in that it has to be overcome by the elevating mechanism, and consequently places an additional load thereon.

THE AUTHOR. Answering Professor Jasper's remarks: The expansion of Formula [4] to the lengthy expression given by Professor Jasper could have readily been included in the code, but after consideration of such a formula when preparing this paper the author decided that since Poisson's ratio does not vary widely for any one material, particularly steel, the simpler relation depending upon $G = \frac{2}{5}E$ and S (in torsion) $= 0.63S$ (in tension) was sufficiently accurate for practical purposes. The paper specifically states that in cases where a high order of accuracy is necessary the exact relation between these various properties should be determined experimentally. Unnecessarily long and complicated mathematical expressions would only tend to defeat the purpose of the paper, namely of simplicity or logical arrangement. Proof of the above relations between the tensile modulus and the torsional modulus, on the one hand, and between the tensile working stress and the torsional working stress, on the other hand, is as follows:

It is safe to say that no matter what type of stress constants, tensile or torsional, are considered, only a definite amount of elastic energy can be obtained from a bar of material. If this is true, then the conversion of $\frac{S^2}{E}$ for tension into $\frac{S^2}{E}$ for torsion in the energy formula [4] should not introduce any factor different from unity. Thus the square of 0.63 is 0.3969, which divided by $\frac{2}{5}$ equals 0.992, which is quite close to unity.

The author does not agree with Professor Jasper that the constant u should be changed to suit dynamical conditions unless the form of the fiber bundle is altered. Instead the value of S should be changed accordingly.

W_m is not always controlled by the elastic limits as Professor Jasper seems to think, since the paper proposes that a "static," "kinetic," or "elevated temperature" material index be used to suit the conditions at hand. In the case of the "kinetic" condition the endurance limit would be the controlling factor.

The author is glad to know that the experimental results obtained by Dr. Brombacher furnish further confirmation of the fact that the old value of the constant k is too high. The only reason that can be advanced at the present time for the author's value

of k being lower than that of Dr. Brombacher is that the former's value corresponds to a maximum deflection, when the error is greatest, while the deflections employed by the latter are probably not so extreme. A large spring concern reports that the old constants are apparently too high, and hence have been designing empirically.

As regards Dr. Brombacher's remark on the use of the relation that " the modulus in torsion is equal to $\frac{2}{5}$ of the value of Young's modulus E, the author has already stated his reasons for using this factor and its limitations in his reply to Professor Jasper.

The author, although fully realizing the importance of the factors hysteresis and after-effect to precision-instrument design, felt that these considerations were details which could be alloted to a separate section in a more complete and expanded code using the present code as a basis. The author touched upon these factors in his previous paper on Mechanical Springs referred to at the beginning of this paper. The subject is also covered by the author in his paper on Overstrain in Metals.[1]

The hysteresis question is brought up again by Mr. Gurney. Apparently increased sensitivity and accuracy is magnifying what heretofore seemed to be an insignificant matter. The difficulties encountered by the Ordnance Department of the U. S. Army in the matter of springs are quite real, thus further emphasizing the urgent need of research in the metal spring art.

[1] Trans., A.I.M.E., vol. 70 (1924), p. 274.

No. 1954

STEEL-FOUNDRY MANAGEMENT

By R. A. Bull,[1] Chicago, Illinois
Non-Member

The author discusses those steel-foundry problems that in his opinion have the greatest general significance in this highly specialized branch of manufacture. He presents a brief analysis of the industry, explaining its technical and commercial divisions, and the significance of each from the standpoint of output, for the purpose of more clearly indicating the nature of the industry under consideration.

Many occupations in the steel foundry are listed in order to explain the complex nature of the operations for which a manager is responsible. The branches of a company organization are assigned to staff and line departments. Reasons are given for certain classifications, regarding which procedures occasionally vary. Policies governing staff departments are discussed separately, not for the purpose of enumerating all duties ordinarily performed, but with the intention of provoking discussion regarding certain factors considered by the author to be of fundamental importance, about which there are differences of opinion.

The subject of compensation both for workmen and for foremen is emphasized because the author believes this is a detail of management which is of very great significance.

Purpose of Paper

THE purpose of this paper is to outline some phases of steel-foundry management which, in the judgment of the author, can be usefully discussed. Convictions regarding details of management are generally the result of experience. The steel-casting industry has become so specialized that the opinions of one steel-foundry manager may materially differ from those of another regarding phases of management which require methods applicable to one branch of the industry, unsuitable for others. The author has endeavored to exclude from his treatment of the subject those details of management that justify marked differences in applica-

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tion, according to the specialized or distinctive nature of the shop and its product.

2 The effort has been made to present general considerations as the result of an experience that has clearly indicated the fallacy of advancing opinions to be accepted without argument. There is no disposition of the author to do more than present his conclusions for what they may be worth as the result of a relatively limited experience to which one individual is necessarily confined.

3 For the consideration of the subject, a preliminary discussion of what a modern steel foundry is, seems advisable. Contact with many engineers has impressed the author with the belief that there is a decided lack of knowledge and appreciation on the part of industrialists, engineers, and scientists concerning the highly complex nature of the modern art of making steel castings. To properly consider the way in which management should function, one must have a fair understanding of what is being managed.

Divisions of the Industry

4 There are several general divisions into which the industry technically separates itself. The first of these is the open-hearth, which is much the largest. Statistics credit the year 1917 with the greatest output of open-hearth-steel castings produced in the United States, viz, 1,357,700 net tons. Of this output, 54 per cent was basic and 46 per cent was acid.

5 The second technical division of the industry from the standpoint of volume of output, is that of the electric furnace. Although electric-steel castings were not produced in the United States until 1908, the amount of such material has reached an annual total of 255,000 tons, approximately 16 per cent of all steel castings produced yearly in this country. Nearly all of the electric-steel castings made in America are of acid metal, presenting in this respect a decided contrast with relation to the fairly equalized selection of acid and basic linings for open-hearth furnaces in the foundry.

6 The third technical division is that of the converter. The amount of steel castings produced from this unit reached the peak in 1918 when 180,000 tons were produced. About that time the output of converter-steel castings was approximately one and one-half times that made from electric furnaces. At present the tonnage of converter castings is about one-fourth that of electric-steel castings.

7 The fourth technical division into which the industry has classified itself is that of the crucible. In 1912, there were 22,000 tons of crucible-steel castings produced in this country. It is doubtful if today any crucible-steel castings are made in America, except in a few instances for home consumption of the product. From the standpoint of commercial distribution, this division can

now be disregarded. The development of the converter for making steel castings, and subsequently, the adaptation of the electric arc furnace for producing steel, relegated to the background the less satisfactory crucible, because of its high cost and restricted capacity.

8 Considered from the standpoint of industrial grouping, steel-casting manufacture divides itself in a different way from that just indicated. Three rather distinct classifications exist, as follows:

(*a*) Open-hearth foundries (generally acid) producing steel castings of sections varying from medium to very heavy, with overall dimensions covering a wide range.

(*b*) Open-hearth foundries (generally basic) delivering a product principally consisting of repetition work in the nature of specialties for railroad use. Such castings are almost invariably of moderate sections, and only a few foundries making this product produce specialties having large overall dimensions. Most of the castings in this class are of average proportions.

(*c*) Electric foundries (almost invariably acid) and converter foundries (always acid), generally specializing in castings of thin sections, and small or medium overall dimensions.

9 Comparing the average size open-hearth foundry with either the electric-steel or converter foundry as commonly found, there are great differences, aside from melting units and other equipment. The distinctions are magnified in the cases of the largest open-hearth foundries and the smallest electric-steel or converter foundries. Many of the characteristics distinguishing each of the two general types are unrelated to the large differences in average tonnages produced in them.

10 It thus will be appreciated that managerial methods should be considered with relation to the type of steel foundry under consideration. In this, as in other industries, there are fundamental principles that can be effectively applied, irrespective of size of output or kind of foundry. Further, there are many details of significance only in the division into which the plant naturally falls.

Variety of Vocations

11 There is no other branch of industry in which the regular operations create as heavy a demand as is made by the steel foundry upon the combined knowledge of the mechanical engineer, the metallurgical engineer, the chemist, the physicist, and the experienced manager of men. One important division of steel-casting work leans heavily on the creations of the expert electrical engineer, to supplement the list of technical and scientific requirements.

12 Many engineers have an inadequate appreciation of the complexities of the steel-casting industry, presenting management problems caused to large extent by the many skilled operations required. A factor of great significance in the manufacture of steel castings is the relative importance of the "personal equation." Labor costs represent approximately half of the total costs of making steel castings.

13 Some persons, including industrialists, distinguish between skilled and semi-skilled occupations by making the following classifications: Trades that are most commonly known to the average man, each of which has relatively large numbers of craftsmen; and occupations, the extent of each of which does not require many workmen, or the nature of which is not familiar to the average citizen who is apt to consider a skilled trade as one that is wholly or partly unionized. The author wishes to make the point that many skilled trades are erroneously regarded as semi-skilled, due to unfamiliarity of many persons with the work involved. It requires much more skill to be a thoroughly competent operator of a pneumatic chipping hammer in a steel foundry than many engineers realize, and probably more skill than is displayed by the average carpenter or bricklayer.

14 The foregoing explanation makes clear the significance of the following list of occupations ordinarily found in the modern steel foundry, requiring men for performance only and not for direction. The complexity of the problems requiring solution by the engineer-manager will be appreciated. Employees such as timekeepers, piece-work checkers, and those doing clerical work only, are excluded from the list:

1 Blacksmiths
2 Blasters
3 Carpenters
4 Casting chasers (For Order Department)
5 Casting checkers (For samples, gaging, etc.)
6 Chippers
7 Coremakers
8 Crane runners
9 Cutting torch men
10 Electricians
11 Grinders (Swing and stand)
12 Laboratory assistants
13 Ladle liners and dryers
14 Machinists
15 Melters and furnace helpers
16 Millwrights
17 Molders
18 Oven tenders (Heat-treating, mold, and core)

19 Patternmakers (Wood and metal)
20 Pattern-storage tenders
21 Sand-mill operators (Facing, backing, and core sand)
22 Scale tenders
23 Sprue cutters
24 Steel pourers
25 Straightening-press men
26 Welders

15 The above occupations are typical in the average large or small steel foundry where patterns are constructed on the ground, where there is a regular gang of maintenance men, and where castings having satisfactory surface appearance are produced.

16 Supplementing the list of typical occupations, we shall enumerate a few important tasks at which additional experienced men must be engaged in the larger open-hearth foundries. These are:

1 Bricklayers
2 Boiler-plant firemen
3 Charging-machine operators
4 Locomotive engineers
5 Locomotive firemen
6 Pipe fitters
7 Stationary engineers
8 Switchmen

17 In any steel-casting plant there are other men employed at occupations, in each of which the work can be done satisfactorily only as the result of careful instruction and considerable experience. Some of these are as follows:

1 Chainmen for cranes
2 Flask (wood and metal) tenders and repairers
3 Gagger and core-rod straighteners
4 Shake-out men
5 Shipping-room attendants
6 Storeroom attendants
7 Tumbling-barrel tenders

Classification of Staff Departments

18 It can be appreciated that the work of running a steel-casting plant should be organized and directed with special care. This leads us to a consideration of the division of responsibilities and the coördination of departments in a steel foundry, operated by men who have headquarters where the product is made.

19 The first group to be considered is what might be termed the staff organization, the members of which report directly to the principal company executives, or to related staff departments. The sections regarded by the author as coming under this general heading are as follows:

1 Sales
2 Accounting
3 Purchasing
4 Cost
5 Drafting
6 Order or Schedule
7 Research
8 Metallurgical
9 Inspection
10 Construction
11 Service

20 There will probably be little disposition to criticize the above listings so far as most of the sections are concerned. Some might disagree with the assignment of certain departments to the group that should be governed by the principal executives. It is appropriate to enumerate reasons for some of the author's preferences.

21 The inspection of the product is a detail that is sometimes given to the superintendent to direct. If the chief inspector reports to the shop superintendent, the author believes a hazard is unwisely assumed that may react to the disadvantage of the consumer, and ultimately to the producer. The up-to-date steel foundry is operated on the principle that the interests of the buyer and the seller are mutual. It is sometimes extremely difficult to decide whether a steel casting should be scrapped or shipped after further treatment. It is impossible to prepare general or specific instructions to cover many points daily requiring decision in the conscientious inspection of steel castings. The proposition largely resolves itself into the exercise of good judgment. For this there often is required a practical knowledge of the use to which the casting would be put.

22 In the necessity for making very fine distinctions between what is good enough to use and what is bad enough to discard, an employee working under the direct authority of the shop superintendent must reflect the latter's judgment in general. He must do what his superior definitely instructs him to do with respect to a particular casting. The superintendent is frequently under pressure to get the largest possible output at the lowest possible cost. Sometimes there is coupled with these requirements, demand for very high quality. The three things cannot always be done at the same time. The desire to do them and thus gratify the management is apt, consciously or unconsciously, to influence the man under fire. The result may be inadequate consideration for the customer's requirements, and for the reputation of the foundry.

23 The only practical means of safeguarding against the exercise of poor judgment in the examination of castings for shipment is, in the author's opinion, to place the chief inspector completely under the jurisdiction of a company executive, who is not as readily

influenced to form judgment from a restricted viewpoint as is the average superintendent.

24 It is frequently advisable to place the chief inspector under the sales manager, when the president or plant manager prefers not to direct this detail. If the sales manager is, as he should be, properly qualified by practical knowledge of shop operations, the author believes he is ideally fitted to supervise the inspection department. It is he who will be most handicapped in the future if the consumer receives unsatisfactory material.

25 The metallurgical department is listed under one of the chief company executives, for the reason that the importance of questions and practices related to the metallurgy of steel are such as to require all the judgment that can be exercised. If the plant manager, to whom appropriately the metallurgical department can report, feels incapable of personally settling a question, he always has recourse to the opinions of his practical shop men. Having these to supplement the sound judgment supposedly characterizing a manager, questions relating to the chemical composition and the physical properties of the metal, and to the details of melting and heat-treatment practice, are most apt to be decided wisely. It is appropriate to state here, that metallurgical problems are much more numerous and frequently more difficult of solution in the steel foundry than they are in the steel mill, when comparable conditions as to variety of metals obtain.

26 It is sometimes found most satisfactory to put the order or schedule department jointly under the direction of the sales manager and the plant manager. The sales department, which sometimes exclusively controls the preparation of foundry schedules, may be influenced too much by the pressure of insistent customers, even though they may have no justification for complaint from the standpoint of commitments. A subordinate in the sales organization who receives repeated and exasperating demands for better service than has been promised or can reasonably be expected, is apt to transmit the pressure to individuals in the shop, who may have to be guided by resulting instructions, regardless of their effect on general operations.

27 It is generally most effective to have under the manager a works engineer to supervise construction and maintenance, to whom the master mechanic may suitably report. This engineer should constantly give attention to power conservation. This includes looking for air leaks, providing, with the least amount of current, the proper amount of artificial light suitably located, the reasonably frequent cleaning of windows and skylights to reduce the required amount of artificial illumination, the systematic upkeep of buildings, and the intelligent direction of all repair work on equipment. For this purpose the works engineer should have access to correct records showing the allocation of maintenance

expense charged to individual items, all of which should be plainly identified in the shop.

Classification of Line or Shop Departments

28 In designating the departments to be headed by the shop superintendent, certain selections ought to be governed by the personal qualifications of the superintendent. In most cases it has been sufficient and satisfactory to give this official jurisdiction over the following departments: Patternmaking, molding, coremaking, flask repairing, melting, pouring, shaking-out, cleaning, heat-treating, and shipping. The innumerable details included in the execution of these duties keep a very industrious and systematic man busily engaged. If the foundry is not of the smallest size, the superintendent should have several assistants. One such should have jurisdiction over both molding and coremaking in any case, and in some cases, patternmaking as well. Another assistant can appropriately be delegated to follow intensively all work in cleaning, heat treating, and shipping. It may be practical to give this latter assistant certain other duties in keeping with his former experience and particular ability.

29 There should be a head molding foreman, the number of whose assistants is governed by the number of molders. The coreroom must have its foreman, and in many cases he needs a subordinate of the type ordinarly termed "straw boss." Cleaning operations include flogging, torch cutting, sprue cutting, chipping, grinding, blasting, tumbling, and welding. Such work includes so many details of importance that one or more foremen may be required to supplement the activities of the delegated aide to the superintendent.

30 Some steel foundries produce sufficient tonnage to require considerable unskilled labor in many departments. For this purpose a "floating gang" is sometimes most suitable. Such a group should be under the control of a general labor boss reporting to the shop superintendent, and in addition, taking instructions from his assistants and sub-department heads. In a fairly large steel foundry, there are innumerable things to be done requiring the cheapest class of labor, and a properly constituted labor gang can invariably be kept busy, conserving the time of better-paid workmen.

31 In shops where night work is necessary, it is generally best to have those in charge of the several crews on the night shift report to one man in general charge, who receives his instructions both from the shop superintendent and from the day sub-department heads.

32 In pointing out that melting should come under the shop superintendent, and that the heat-treatment ovens should come under one of his assistants, reservation is made regarding orders

to govern manipulation, transmitted by those higher in the organization, for the execution of which the shop men mentioned should be responsible.

33 It is superfluous to outline many reasons for assigning certain shop responsibilities to designated individuals. The arrangements suggested for the work to be done by the line rather than the staff departments are not submitted with the idea that they will suit all conditions. The general scheme outlined has worked well in certain foundries of average size and character. There is, however, a wide range of desirable modifications of any shop-department plan because of enormous differences in nature and tonnage of plant output. Hence, there is more opportunity for fruitful discussion of suitable arrangement of staff divisions than there is of line departments.

34 A foundry such as the author has been considering is one that, so far as the company officials are concerned, is self-contained; that is, one operated as a single unit, and not one of several foundries in a consolidation.

Branch Plant Management

35 Obviously where several plants are owned by one company, procedures must be adopted for a discriminating degree of authority given to local directing representatives. The author's experience indicates that the resident manager should be given reasonable freedom in prescribing how specified results may be obtained. He should have no restrictions placed on the selection of his subordinates. He should have a voice in decisions regarding any operating procedures laid down to be strictly observed. His judgment ought to be followed to a large extent as to policies governing relations with workingmen, because of his familiarity with local conditions. Sometimes a large corporation expects results from the resident manager without giving him proper latitude in following his own judgment. The man in charge of a plant detached from headquarters occasionally occupies an unpleasant position, the difficulties of which are inadequately appreciated by his superiors.

Policies Governing Staff Departments

36 Some details can appropriately be mentioned regarding the manner in which the staff departments should function, although the nature of the particular foundry's business has much to do with such matters. It is possible to outline some of the major considerations, from a general standpoint.

37 *Sales Department.* A far-sighted merchandising division in any branch of industry should not generally expend useful effort in attempting to develop business that from economic standpoints belongs elsewhere, either because of other material better suited

for the job, or because of the location of producer with respect to consumer. It may be wise to make a temporary exception in an emergency, such as a very low volume of suitable business available. No sales department should try regularly to secure orders for parts that will be entirely satisfactory when produced from weaker and less expensive metals. A well-managed steel foundry ought to first clearly understand its proper function in industry. The fitness of the part for the use intended should always be carefully considered. Any other policy, if carried sufficiently far, will react unfavorably on the management responsible, and in the end will benefit nobody.

38 It is necessary, in a far-sighted merchandising policy, to have the closest coöperation between sales and operating officials. The latter, because of early training and present responsibilities, have their practical sense well developed. They often have sound ideas as to the ultimate result to the consumer and to the steel foundry, of efforts to secure business in certain fields. Sometimes the attitude of the operating head is that of a disgruntled man who feels under the handicap of making, against his protest, work that is entirely unsuited to his equipment or that belongs by every economical consideration in some other branch of the metal industry than the steel foundry.

39 The sales department can usefully employ a man, and sometimes several, much of whose time can be devoted to "service." To be effective, this sub-department must have the combined viewpoint of the operating and selling organizations. It should be under the direction of the merchandising department. Every one in this sub-division should be thoroughly familiar with the way castings are made, and therefore be able to explain to a consumer the reason for any defects regarding which he may complain. Sometimes a consumer's attitude towards an imperfect product is completely changed after learning of the numerous factors involved and the vast amount of effort expended in doing a perfect job. It never pays to practice the slightest deception in such a matter. The customer has a right to know the facts.

40 *Accounting Department.* The accounting department of any relatively large plant sometimes acquires systems involving costly records that may continue to cause expense long after they serve useful purposes. This is liable occasionally to occur without the realization of the management or the men supposed to be aided by examination of the data.

41 It seems advisable for an executive periodically to make a comprehensive analysis of all clerical records, and regulate their continuance by their future utility. In some cases there will thus be disclosed a good use, now discontinued, that was once made of a record, that could serve a useful purpose. In other cases clerical work is performed at considerable cost daily to prepare

records that are mechanically distributed and filed by the man supposed to be helped by the records, but who, through the development of the business or department, no longer has need for them. It may be unfair to blame a department head for failing to immediately call attention to the fact that he no longer requires forms that were previously useful. A man in charge of many shop details generally has his mind on more important matters than that of the real utility of all report forms.

42 Campaigns to eliminate superfluous records and to dispense with unnecessary clerical help have in certain foundries materially reduced overhead expense without imposing a handicap of any kind.

43 *Purchasing Department.* It is unnecessary to make extended reference regarding the purchasing department. However, it may be pointed out that there are some buyers to whom the management gives more latitude in the selection of materials on a price basis than is economical. This evil has prevailed in the steel-foundry industry as in others. Men responsible for mechanical operations are entitled to the full consideration of their views regarding the suitability of commodities and equipment.

44 One large steel-casting plant whose purchasing department was conducted very efficiently, apparently permitted all selections to be made by the purchasing agent, while in reality the procedure was far different. The buyer was guided strictly by the preferences of the operating department as given to him after a study of all proposals, gaged in degree of desirability expressed whenever possible, in definite terms. For example, if the foundry needed a lathe for its maintenance department, the operating head requested the purchasing agent to secure proposals from sources indicated by the plant. A careful examination of the good points of each tool offered was made, followed by a report to the purchasing department indicating, first, second, and third choices, and the extent to which each choice was justified in first cost according to the judgment of the plant. This scheme was followed with perfect satisfaction to the two departments concerned.

45 *Cost Department.* The steel-foundry industry has probably suffered as much from lack of attention to cost finding as has any other manufacturing business. It has undoubtedly needed improved methods of cost accounting more than have numerous other industries which, because of their product, more easily permit the installation of a good cost system. The jobbing foundry presents a difficult problem in the adoption of many cost-finding schemes that can be satisfactorily applied elsewhere. The principal element that must be kept in mind in applying any system to a steel foundry is related to the sense of proportion. Some schemes are too elaborate to justify the expense of determining actual costs in view of the returns on a small order for castings.

46 In the foundry making a wide variety of castings on orders, each of which frequently calls for very few pieces, the general procedure in attempting to solve the cost problem is to classify by weight. In the earlier days of the industry weight classification was of undoubted benefit. It was the first effective step that could have been taken.

47 The weight of a casting generally has a great influence on its cost of production, and consequently on its proper selling price. Other factors, however, make weight classification alone very deceptive. It is unfortunate that, up to the present time, many steel founders have adhered to the practice of classifying foundry costs by weight divisions. This has handicapped the industry and all served by it.

48 There has been considerable discussion of more effective and thoroughly practical plans of ascertaining costs in the steel foundry. A paper presented by W. J. Corbett at the convention of the American Foundrymen's Association in Milwaukee, in October, 1924, occasioned a large amount of constructive comment. A study of the scheme outlined in that paper is suggested to those particularly interested in this subject.

49 *Drafting Department.* The amount of drafting work necessary in a steel foundry may be so small as not to require the entire time of one person.

50 Every steel foundry doing jobbing work must frequently calculate estimated weights from customers' drawings. This is properly the draftman's job, and sometimes requires much time. Many inquiries are received for prices on castings, the weights of which are not indicated on the customer's drawings or requests for quotations. Engineers associated with companies consuming steel castings can do much to help themselves and assist the foundry by showing the estimated or actual weights of castings on all drawings. The merchandising department is unable to intelligently establish all important factors relating to the appropriate selling price until the weights of the castings are definitely known or estimated.

51 Failure of the customer to furnish weights of castings previously made at another foundry is generally due to oversight by the purchasing agent. The omission of estimated weights on a drawing for a new casting is frequently an unjustified error, due to lack of understanding of weight significance to the foundryman. In most cases the engineer is required to estimate the weights of new castings in order to ascertain the probable cost of the equipment of which the castings are to be parts. The prospective purchaser usually has the information, to give to those requested to bid.

52 All foundrymen would appreciate a more pronounced disposition by engineers to keep this detail in mind. Better coöperation between purchaser and consumer along this line will assist

the prompt receipt of quotations in closer conformity with production costs.

53 *Order or Schedule Department.* Specific reference should be made to a few details in the order or schedule department. Some shops have more or less novel and effective adaptations of the general plan for conducting this small but important division. One consists in a record of the number of delivery promises kept during each month. The final percentages are scrutinized with care by the management. Such a plan probably brings the best judgment to bear on the customer's request for information as to shipment.

54 In some foundries daily meetings are held, presided over by the head of the order department, and attended by every shop foreman concerned in the execution of orders for castings. All requests for delivery promises are there considered, and information is exchanged regarding the elements involved in delivery. The plan undoubtedly has some advantages. Its application in any foundry should be considered with reference to the practicability of assembling all interested persons at one time, thus simultaneously removing from their departments the men concerned.

55 *Research Department.* The research, or experimental, department should be a very active one, although its personnel may sometimes be a single individual. When one man is selected to direct all of the investigations, provision must be made for periodically giving him assistance. This generally causes no complications to routine operations. The extent of the plant's activities may easily be such as to justify regularly the time of several persons for experimental work. The man actively in charge of research should report to the plant manager and be in close contact with him. A manager generally has a much more practical idea of results that may come from investigations than has an investigator, who because of his regular occupation may be absorbed in a problem that appeals more to the imagination than is justified by its probable useful application.

56 The research engineer, except in rare cases taking little of his time, should not have responsibilities relating to production routine. The smallest steel foundry gives ample opportunity for one good man to spend all his efforts on technical investigations that will improve quality or reduce costs.

57 *Metallurgical Department.* The metallurgical department of a well-organized steel foundry should be the source of general instructions to those in charge of melting and heat-treatment. It should scrutinize current literature relating to these operations, and to all matters connected with tests of significance to the industry. The company should keep informed of actual or needed developments in formulating and revising specifications for its product.

58 A well-qualified metallurgist is a highly valuable member of the organization. His duties should include much more than the general supervision of routine furnace operations, physical tests, and chemical analyses. He should collaborate with the research engineer, and like him, should have a practical point of view to balance his scientific training.

59 *Inspection Department.* In the regulation of a systematized inspection department, much depends on the suitability of the man in charge. His task is daily associated with as many complexities and demands for the exercise of good judgment as is that of the man in charge of the order department. This individual probably should have more disposition to act on his own judgment alone than is advisable in the case of many other subordinates in the steel foundry, in order to successfully and conscientiously discharge the duties outlined, with regard for the customer's interests. Definiteness of conviction should be preceded by much practical knowledge of foundry conditions and by considerable information regarding general industrial application.

60 A casting may be suitable from the standpoint of surface appearance, but contain vital defects in the interior. The chief inspector should have facilities for ascertaining the soundness of sections, to remove all doubt before the product is delivered. This may be satisfactorily accomplished by systematically destroying representative castings by machining operations that disclose the interior. The most skilful foundrymen cannot be trusted always to produce castings free from shrinkage cavities and other serious defects, unless aided by observing samples cut into pieces. When this discloses an undesirable condition, the experienced foundryman can modify his practice to overcome the defect.

61 The use of templets and gages by the chief inspector's subordinates is made very satisfactory by carefully checking these with master gages in the execution of production jobs. The importance of this is not always realized. Gages are sometimes zealously used, but are ineffective because of wear. If the nature of the order for castings makes master gages impracticable, there should be periodic checking of regular gages from drawings.

Compensation of Workmen

62 In large foundries it is advisable, in the author's opinion, for the plant manager to have a man who has had considerable experience in the operating departments, who has tact and cleverness, and who combines carefulness with proper consideration for the mutual interests of the men and the management, to be continuously engaged in the study of compensation to workmen doing piece work, task work, and day work.

63 Too many employers have in the past pinned their faith to the installation of a piece-work plan as the cure for their troubles with workers. Much thought should be given to the

operation of a piece-work scheme. Probably in no other industrial enterprise is this more true than in the case of the jobbing foundry, because of the great variety and short duration of the tasks, some of which are of such nature as to preclude satisfactory piece-work rates.

64 There are many more or less ingenious plans for applying piece-work principles to steel-foundry operations. Some guarantee a minimum daily wage to the man, and give him in addition, a reasonable sum for a stated amount over the established number of tasks. Certain compensation plans go further, and provide for increasingly greater piece-work rates as the speed of the man advances beyond the average rapidity of execution. Some plants pay bonuses semimonthly or monthly for constant attendance during designated working hours. Some production jobs that have been running in foundries for long periods can profitably have applied to them bonus schemes, based on a minimum percentage of defectives.

65 Piece-work compensation of molders has in some foundries been advantageously developed by means of time studies to show the proper relative significance of every detail of a molder's task. Such a scheme employs a unit system for calculating equitable rates, by indicating the time required for each step in making a complete mold. Such a system, carefully planned in a spirit of fairness, possesses important advantages, particularly in establishing without delay, suitable rates for new jobs, and in convincing molders, as the result of experience, that they are getting a square deal.

66 There are certain well-intended compensation schemes for workmen which are thoroughly equitable in principle and which sometimes appeal to inexperienced executives, but which result in relative inefficiency because of their complicated nature. The average workman in a shop where operations are either repetitive or of a character producing more or less of what the inexperienced would call confusion, is generally of a type unable to satisfactorily grasp all the details of an elaborate wage scheme involving numerous factors. A plan that is perfectly clear to educated men will often fail completely to be understood by those who have not had the advantage of much mental training. This is true of workmen of any nationality or color.

67 It is well to daily advise men who do piece work and who are paid bonuses for low percentages of defectives, regarding the results of their work. Such advice is sometimes given in the form of posted figures on the man's floor or bench where he and his associates can examine them. This occasionally produces a wholesome rivalry that is more stimulating than the increased wages.

68 It is appropriate to say in passing that there are certain operations in the steel foundry where piece work on a tonnage or other basis should not prevail, for reasons that relate to quality.

69 Given suitable equipment and the proper amount of direction, experienced molders will do satisfactory work at a stated price per mold. In this and similar foundry operations there is not the danger of sacrificing quality for quantity that exists, for example, in melting steel, in heat treatment, or in welding castings. Men who are paid for quantity production in the operations just mentioned and in similar ones are prone to be influenced in their work in a way to detract from the quality desired by the management.

70 Where piece work cannot be effectively employed, a task-work plan is sometimes advisable. Much thought should be given this arrangement as applied to a crew of men, to prevent abuse. The inherent weakness of the scheme as ordinarily employed lies in compensating each man in the gang on the same basis, despite the fact that some individuals may be much faster workmen than others. The larger the crew, the greater will be the difference in the work done by each person, with resulting justified dissatisfaction.

Compensation of Foremen

71 The compensation of those actually responsible for directing shop operation, beginning with the superintendent and including such men as core foremen and cleaning-room foremen, requires as careful consideration as does the compensation of the workmen. Many steel foundries supplement the monthly salaries given these men by bonuses for individual meritorious achievement. Certain of these bonus arrangements are extremely interesting both as to plan and apparent results. Without doubt some sort of bonus plan is desirable for the payment of foremen, provided fundamental considerations are satisfied. These include first, the payment of a salary fully in keeping with the routine responsibilities carried, and with local conditions (if these in any degree establish a rate for a similar occupation); second, a bonus that will reward the faithful and effective individual liberally for signal service performed; third, a bonus scheme, all details of which can be fully explained and completely understood by the men to whom it is applied; fourth, a plan which, in justice to the management, is based partly on the general financial showing of the plant or on factors that cannot be dissociated from it, to prevent in hard times the payment of compensations on a scale out of proportion to prevailing circumstances. It is extremely undesirable to reduce salaries except under the most exasperating or critical conditions. It is also inadvisable to temporarily abolish a foremen's bonus plan, or modify it so that it will carry smaller perquisites than were originally prescribed.

72 A bonus plan for foremen should, therefore, be rather comprehensive and be developed with ample consideration to operating profits and the man's part in creating them. When substantial

bonuses are earned by a few principal subordinates at times when the company's operations as a whole are not profitable, payment should be made gracefully as long as it is made at all, so that the good effect of the reward on the recipient will not be nullified.

73 Any plan for paying bonuses to principal shop subordinates should provide in an auxiliary way for each of a number of valuable men performing duties not directly related to production routine. If such men are not included in the bonus plan, they invariably feel that circumstances have unfairly operated to prevent their earning appropriate compensations for the fidelity and skill with which they may perform allotted tasks. It is often difficult to establish a special compensation plan for each of the men in this general class. The management should so solve the problem as to convince employees that they will be specially rewarded for particularly meritorious execution of duties.

Conclusion

74 Responsibility for the direction of any industrial enterprise is apt to make those in charge feel that their particular problems are more difficult to solve than those of many other branches of industry. If this outline of the author's observations and opinions regarding some phases of steel-foundry management should give engineers engaged in other activities a clearer idea of the details of steel-casting manufacture, the result may be closer coöperation between some who make, and others who use, the product of this important factor in modern civilization now capable of supplying 2,000,000 tons of castings annually, from a total of 350 steel foundries in the United States. This giant of industry has acquired all of its growth in America during less than 60 years. The remarkable development thus indicated has been accompanied by a progressive increase of technical and commercial problems requiring solution by the steel-foundry manager.

75 The author realizes that important phases of the subject have had inadequate treatment in this paper. He hopes for charitable consideration for omissions, and for the unsoundness of any of his expressed opinions, none of which have been arbitrarily advanced. Criticisms of any of these will be appreciatively received.

DISCUSSION

A. W. Gregg.[1] The author has laid down general principles of steel foundry management with which no one can disagree. However, some disagreement may be permissible as to details, particularly where the work is of varied character. Three points on which the writer disagrees are the order of schedule, the metallurgical department, and construction. The author has these de-

[1] Bucyrus Company, South Milwaukee, Wis.

partments reporting to officials of the company. The writer believes them to be functions of the operating department. He believes that the order of schedule that can be best worked out by the operating department is the proper one, and that metallurgical work in any plant is also an operative function.

In regard to inspection, in the company with which the writer is connected, the chief inspector reports to the works manager. The writer, however, believes that the author is correct in his statement that a hazard is introduced when any man who is interested in costs has the final say on inspection. He will perhaps, in spite of himself, be unconsciously influenced by the fact that he is under pressure for production, and costs and quality should be entirely aside from that. The man passing on quality should have no interest in tonnage produced, or the cost of it.

A. B. Segur.[1] The author has made special reference to the use of stop-watch time studies to determine the time allowances for piece rates. He also made particular reference to the value of determining standards and units by which the cost of castings might be calculated. He referred further to the general idea of forming a unit of measurement whereby one series of time studies might be made to serve over a fairly wide range of castings or of flasks.

It is doubtful if there is an adequate conception of the possibilities along this particular line, for instance, in flask molding and coremaking, which are big factors in any foundry.

Some years ago Mr. Frank Gilbreth made about 65,000 observations on the fundamental time required to perform basic operations in connection with coremaking and with flask molding. He obtained accurately the time required for a man to strike a blow with a hammer or to strike a blow with the ram in ramming up a mold under a wide range of conditions; the time for making one turn of a screw in clamping the flask; the time required for all of the ordinary operations which would be involved in any ordinary flask molding or in any ordinary core job in the same way; and the proper method of proceeding on each job.

The object of this investigation was to find how an expert would proceed on a particular type of coremaking and flask molding. This expert was under pressure to do the best that he could do on that particular job, especially because the work was done for the benefit of the blinded soldiers, sailors, and marines in the United States forces, at a time when it was thought there would be some 40,000 of these men.

As he performed the operation he was photographed with a high-speed motion-picture camera and in the field of the picture was placed a clock which measured time to 0.0005 min. When he

[1] Oak Park, Ill. Assoc-Mem. A.S.M.E.

obtained the time required to strike one blow with the hammer, the accuracy of that time was unquestioned. All the conditions which surrounded the striking of that blow with the ram or hammer could be reproduced at any time, and the reasons for discrepancies could be easily ascertained.

At the same time studies were made of a number of other operations, including furniture finishing. In furniture-finishing operations, the question resolves itself into time required for a man to move his hand one foot under various load conditions. In a study of some thousands of these observations it became apparent to any one who was acquainted with a particular type of job in furniture finishing that the time required for a man to move his hand one foot would not vary between operators by an amount to exceed 0.0005 min., and that a variation from the average did not occur one time out of a hundred. In other words, in 99 per cent of the observations there was no difference whatever between operators in the time required to perform this fundamental operation. Also, on the other operations, outside of the time required for the actual movement of the hand, there was the same degree of accuracy in time between operators. There was a vast difference between operators, but the difference was not in the hands but in the head. One operator, when he had finished moving his hand one foot, knew what to do next; the other operator did not. Consequently, when it came to the total time required to perform a given task the more skilled operator had by far the greater advantage in time, although the effort put forth and the time required for a single stroke was exactly the same.

In the application of this information to furniture finishing we find that it is now possible to take any piece of furniture, measure it with a two-foot rule, apply the fundamental time standards to the measurement obtained, and thus obtain a standard piece-rate time allowance which is far more accurate than the stop watch is able to record.

The time required to obtain a proper allowance under circumstances of that kind is negligible as compared to the time required in an accurate stop-watch study. The cost is not over ten per cent. In applying that method to the furniture industry we are able to determine accurately losses that are occurring both in quality and in quantity of work performed that would be absolutely impossible under stop-watch methods.

The only reason for using furniture as an example is to bring out the possible application of the same method to coremaking and flask molding. Now for flask molding 65,000 observations have been made and analyzed. The time required for performing fundamental operations, such as the time required for making one stroke with the ram where there were different operators, some men and some women, of different builds, working on different types of flasks, and different kinds of cores and molds entirely, varied by

less than 0.0005 min., and the average variation was less than 0.0001 min. between different operators. The same results have been secured from very careful studies of hundreds of observations and hundreds of tests. Other operations involved were similar and had just as great accuracy.

Studies at the present time would seem to indicate that, with proper study, coremaking and flask molding might be just as thoroughly standardized as furniture finishing. On the same type of cores and within certain measurements, preliminary studies would seem to indicate that the unit of measurement bears a very close relation to the size of the piece being packed or rammed.

If in the foundry it were possible to obtain such a unit, then the question as to what the proper unit of molding should be would be clearly defined. It would not take very long to evolve a sales unit, by which castings could be sold, that would conform more nearly to the cost of manufacture than do the ordinary units now in use.

Furthermore, in every foundry that is using piece rates, the men have among them a tradition that one job is easy and another is difficult. There is always much dissatisfaction among the men about the rates that are set on different jobs. This dissatisfaction may take the form of suspicion, open rebellion, or simple acquiescence. In very few cases it may be felt that all the rates set are correct as set. However, it may be possible to establish a standard based on the number of units of work involved, which standard both the men and the management will easily understand, and which will prevent these questions from arising. The units of work can be put in a form that will not contain the technicalities that the writer has introduced into the method of obtaining these units of work.

In the next place, if it is possible to obtain an absolute standard by some such method of measurements, we have at once the thing that every engineer, every manager, wants in the determination of whether the methods used in his foundry are correct, because if a particular job is performed within the time standard, based on the number of units of work involved in it, there is no longer need to worry about that particular flask or that particular core. If, however, some other job is considerably off standard, it is at once known that the method is probably wrong on that particular job and the production problem can at once be attacked at the point which requires attention.

Furthermore, once the time required to perform a job is determined for a particular method of getting quality, then there is established a standard procedure for obtaining quality impossible to obtain by any other means. This is so because both man and management then know something definite on which they are to base the practice in that particular foundry. Quality will not suffer provided methods are properly followed up.

In the writer's opinion the reason that the piece rate falls down in so many instances is because there is a constant suspicion on both the part of the man and the management that the original basis of these piece rates was not correct. Each man, then, is trying to take advantage of the other to get what he thinks is his due. When both sides can come together on some common-sense means of determining what is correct and will work out, then we shall have reached the stage in foundry management that the greatest of all problems, the human factor, will have come close to a practical solution.

A. E. HARRISON.[1] In the Allis-Chalmers Company the chief inspector serves nine departments, of which the foundry is one, and reports to the general works manager, but the latter rarely interferes with the work of the inspector. The inspector is almost supreme in his department, and if any difference of opinion arises the superintendent of the department that questions the ruling of the inspector immediately calls in the manager of the department where the work originates and he himself either serves or appoints one of his engineers, which forms a committee of three, consisting of the chief inspector, the superintendent who has questioned the inspector's decision, and the representative of the manager of the department. The ruling of these three on this particular work is final.

This procedure protects the works manager from having to take sides in detail matters, and compels the engineering department to work with the inspector to rectify conditions that tend to produce poor castings. It is not necessary that the inspector report to the works manager in every plant. He could just as well report to the president or some other high executive. In the Allis-Chalmers organization, however, it is more convenient to arrange matters as above outlined. The important consideration is that the inspector be given a free hand and that he be required to work with, not under, department heads.

CHARLES J. SIEGLE.[2] The writer believes that in the interest of a casting the management should call in both the engineer and the foundryman as well as the inspector. Castings are frequently produced that have the appearance of being poor castings, yet when they have been machined they have proved to be perfectly sound, even though they had previously been rejected by the inspection department. No one but the man that pours it really knows what a casting is, and what the inside of it is like. If a disturbance occurs in a mold, the foundryman knows that something has happened, while the inspector does not. The inspector pits his

[1] Allis-Chalmers Mfg. Co., Milwaukee, Wis.
[2] Harnischfeger Corpn., Hercules Plant, Milwaukee, Wis.

judgment against that of the foundryman and condemns the casting. The writer has been on committees that have discussed this question, and has found both the inspector and the engineer to be wrong. They have refused to accept the foundryman's word that the casting has been poured properly, and have rejected castings that had a few surface defects, which afterward have proved to be sound.

The proper procedure would seem to be first to consult the foundryman that poured the casting and ascertain his opinion as to whether or not something has occurred to cause an apparent defect. If the foundryman could give no information, then it would be a proper subject for discussion by the management and the engineering department. So long as a mold is poured properly, the foundryman should have a voice in the rejection of a casting. If a disturbance occurs in pouring, then there is something on which to base a search for defects. If one does not know of the disturbance, any search then is simply guesswork. No man can look inside of a casting and condemn it unless he knows how it was poured.

NAND SINGH.[1] It would be of interest if the author could enlighten us on the staff organization of a foundry, and on the trend of modern management in foundries, and also on the training or prerequisites that students should have before they enter foundry work, after they have completed their school or college course.

We all understand the principles on which any organization is based. We have as the various types of organization, the functional, the line and staff, and finally the scientific organization as first developed by Taylor. It might be pertinent to inquire as to whether the author has been able to find any place where these types of organization have been differentiated, so that in training of students who probably will be managers in the future, we could keep in view something of each type of organization for the purpose of training the student in its fundamentals.

THE AUTHOR. The point concerning the inspection probably was not clearly understood by some of those who discussed the paper. It was pointed out in the paper that the inspection ought to be under a company executive and not necessarily under the sales department. Reference to Par. 22 may make this clear. In regard to the point that was made covering the pouring of the casting, the author believes that one of the functions of the right kind of a chief inspector is to see all heats poured. It was one of his duties when he was inspecting castings. In other shops with which he has been acquainted the inspectors were qualified, by

[1] Professor of Business Administration, Marquette University, Milwaukee, Wis. Mem. A.S.M.E.

seeing the heats poured, to tell whether or not the molds were properly cast.

Mr. Gregg takes issue with the placing of certain departments under the company executive instead of under the operating department. Some, perhaps unconsciously, might make a distinction there that was not in the author's mind. The head of the operating department should in every sense be a company executive, and when the fact is stressed that company executives ought to have the direction of certain departments, it is not meant that the heads of the operating department should have no direction of those matters.

The author's conception of an ideally constituted steel foundry is for a company executive of commanding relative position to be the head of the operating department. That might remove some of the questions regarding the placing of the metallurgical, schedule, and construction and maintenance departments under one man as compared with another man.

Concerning the special training which is necessary to equip young men to hold executive positions in the foundry, it would be very difficult for anybody to be very specific, or to say that particular training along any special line (of course assuming that engineering training is highly desirable in the first place) is more apt to produce the best type of executive. The question probably is related more to aggressive personality than to any other one thing. The best kind of foundry executives, or the best kind of industrial executives, generally, are those men who have been blessed by nature with the strong and pleasing personalities, combined with the necessary amount of force and aggressiveness that permits them to exercise their personality in an effective way. No other asset is so valuable in the direction of a foundry and perhaps in any kind of industrial enterprise. Such things cannot be put into a college curriculum, but they can be cultivated at the age when young men are in college, through the medium of close intercourse between the professors and the young men over whom they may have charge.

Of course a reasonable knowledge of accounting as well as a broad knowledge of engineering work and considerable knowledge of metallurgical work, and a study of merchandising methods, are all highly valuable to a man after he finally obtains a managerial position at the head of any large foundry organization. But if the man is gifted by nature with the kind of a personality that will draw men to him and induce men to carry out his instructions, he is perhaps better equipped than if he was particularly proficient along any other one of those different lines.

No. 1955

BOILER FURNACES FOR PULVERIZED COAL

By A. G. Christie,[1] Baltimore, Md.
Member of the Society

Powdered coal is used more widely every year for furnaces under boilers. An analysis of the fundamentals of the combustion of the coal and of the effect of radiant heat and flame on the performance of such furnaces is presented in this paper. Certain features secondary to the furnace and dealing with preparing and drying coal and handling ash are also discussed briefly.

This analysis indicates that the following conditions are desirable for the highest efficiency and maximum capacity of a given furnace. Coal should be finely ground, thoroughly dry, and preheated before entering the furnace. Primary and secondary air should be highly preheated. Turbulence should exist inside the furnace. The walls should be water-cooled. New methods of handling ash and preventing the discharge of dust into the air should be developed. These will lead toward greater capacity for a given furnace volume and to higher boiler efficiencies.

THIS paper will consider some fundamental principles in connection with the combustion of pulverized coal and will note how these may be applied in the design of suitable furnaces. Certain methods of preparing, drying, and firing coal and of handling ash will also be briefly discussed.

Methods of Heat Transfer

2 A clear conception of the methods of heat transfer is necessary for an understanding of furnace problems. There are three methods of transfer, viz., conduction, convection, and radiation.

3 Conduction is the intermolecular transfer of heat within a body. For instance, one end of a copper bar is placed in a flame. This causes a large increase of molecular activity which is known as "increase in temperature." The vibrations of adjacent molecules increase and in this way the activity proceeds from molecule to

[1] Professor of Mechanical Engineering, Johns Hopkins University.

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molecule along the bar toward the cold end. If one end is kept cold, heat will continue to flow from the hot to the cold end. This physical phenomenon is called "conduction." Heat is transmitted through the metal walls of boiler tubes in this manner.

4 Convection refers to the mass transfer of heat. For instance, a certain volume of gas is heated by contact with a hot surface or by combustion, and rises due to decreased density or draft. Later the gas comes in contact with a cold body, to which it gives up its heat. A greater part of the heat is delivered by flue gases to the second and third passes of a boiler, to economizers, and to air preheaters by means of convection.

5 Radiant energy is a form of activity in the ether similar to light and electricity. Heat radiations are wave actions and differ from light and electricity only in wave length. The molecules of a hot body are assumed to be in that violent state of vibration or oscillation associated with the hot state. These molecules set up radiant-energy waves in the ether which travel at enormous speeds in straight lines in all directions from the hot body. When such radiant-energy waves impinge on a colder body some of whose molecules can vibrate in tune with them, then the waves are absorbed by the colder body. Its temperature rises due to the increased vibration of its molecules resulting from the absorption of the radiant-energy waves. Thus radiant energy passes to all boiler surfaces directly exposed to the fire and is conducted through the metal to the boiling water inside. If the body contains only molecules which do not have natural periods of vibration in harmony with the impinging waves, the radiant energy either passes through or is reflected. Thus heat waves pass through glass without appreciably raising its temperature. On the other hand, many insulating materials such as firebrick reflect the radiant-energy waves from their surfaces and absorb little heat.

6 A dull black body has the greatest property of emitting and receiving radiant energy. A gray body, on the other hand, emits and absorbs only a small portion of the energy of a black body under similar temperature conditions. In order that heating surfaces may absorb radiant energy they must "see" the source of the radiations. This is important in adjusting the areas of boiler surface exposed to the furnace.

7 Certain gases also have the property of radiating heat. For instance, carbon dioxide and water vapor can both absorb and emit radiant-energy waves. Oxygen, nitrogen, and air, on the other hand, are practically inert as radiators or absorbers. Hence heat radiation from the gases plays an important part in the performance of a furnace. It is therefore necessary to study closely the laws of radiant heat as well as those of convection as applied to furnaces.

8 Stefan and Boltzmann's law of radiant-heat transfer states that for an ideal black body the amount of heat transferred varies

as the difference between the fourth powers of the absolute temperatures of the hot and cold bodies. This takes the form $H = K(T_1^4 - T_2^4)$, where H is the heat transferred in unit time, K a constant depending on the nature of the surfaces of the hot and cold bodies, and T_1 and T_2 the absolute temperatures of the hot and cold bodies, respectively. Actual solids and gases follow this law more or less closely. The rapid increase or decrease of heat transfer with relatively small changes in furnace temperatures is the significant feature of this formula. Experience with furnaces has shown that it is desirable to transfer as much heat as possible by radiant energy. Hence from this standpoint, maximum furnace temperatures are desirable. Any influence which tends to lower furnace temperature should therefore be closely scrutinized and, if possible, removed so that this desired maximum temperature may be secured. This idea must be kept clearly in mind in considering designs of furnaces for pulverized coal.

9 The available knowledge of heat-radiation phenomena from solid bodies and from gases and of conditions governing receiving surfaces, is very incomplete. It is surprising in view of its great importance that radiant energy has been given so little consideration in the literature of thermodynamics, while heat transfer by convection has been studied extensively. In fact, it may be later discovered that some of the contradictory results found in the study of convection may be due to radiations from the gases used in the experiments. The true significance of radiant heat in furnace design has only been recognized within recent years. Increased boiler efficiencies have resulted from the design of furnaces which utilize radiant-heat transfer to a greater extent than in former practice and further improvements are still possible, some of which will be discussed.

Nature of Combustion

10 The elements entering into the chemical reaction known as combustion are those occurring in the coal together with the oxygen of the air. The elements in the coal as given by an ultimate analysis are carbon, hydrogen, oxygen, nitrogen, sulphur, and ash. Little is definitely known regarding the association of the carbon, hydrogen, and oxygen in the coal, particularly in bituminous and lignite coals. Certain destructive changes involving distillation take place in the coal when subjected to heat, and as a result hydrogen, oxygen, and the so-called volatile carbon in various combinations are evolved. Furthermore these distillation changes take place and the resultant gases are burned before the remaining carbon of the coal, now in coke form, is consumed. Sulphur may occur as free sulphur or as pyrites, frequently the latter. Sulphur dioxide is formed by oxidation of pyrites and the free iron combines with oxygen to form Fe_2O_3. This iron oxide enters the ash and,

as will be shown later, has a tremendous influence on the life and performance of the furnace walls.

11 The most rapid ignition and combustion possible is desired to obtain the highest furnace temperatures and thus the highest efficiency. The small powdered-coal particle entering the furnace at the burner mouth must first be heated up before combustion starts. This heating is by stages. First, any moisture in the coal must be vaporized. The drying process is endothermic or heat-absorbing. That is, the furnace walls or the burning flame must furnish this heat to raise the temperature of the incoming coal particle and to start distillation, which is the next stage. While some of this radiant heat is furnished by the surrounding walls, the greater part comes from the flames of burning coal and gases. This reverberatory action of the flames is highly important, particularly with low-volatile coals. The distillation changes start at about 400 deg. fahr., and gases and volatile vapors are given off forming smoke until combustion starts. Experiment indicates that the distillation process is also endothermic unless the oxygen content of the coal is high.[1] The entering coal particle, being a black body, readily absorbs radiant heat and increases in temperature. The primary air which accompanies the coal is a poor absorber of radiant heat and may have increased in temperature only slightly when the gases begin to distill. In fact, it is quite probable that the air is heated largely by convection from the coal particles suspended in it and which are absorbing radiant heat. The smoky gases from the coal contain carbon particles which continue to absorb radiant heat, thus increasing their temperature. Finally the air-gas mixture reaches ignition temperature and combustion of the gases commences.

12 The preceding paragraph will make it evident that the coal particle must have progressed some distance into the furnace before combustion commences. Observation of furnace conditions confirm this fact. A portion of the flame travel in the furnace must therefore be charged to this heating stage. It would be desirable to have ignition occur earlier in order to reduce flame travel and consequently furnace volume. This end may be approached by having the entering coal as dry and hot as possible and accompanied by highly preheated primary air. Driers have not been installed in all plants and it has been possible to operate with certain coals without driers. However, to get maximum results out of properly designed furnaces, well-dried coal is an essential as indicated by the previous analysis.

13 The primary air now used to convey coal into the furnaces is often preheated to some extent, but rarely over 150 deg. fahr. This air is the best means for heating the entering coal and should therefore be highly preheated. This preheating will obviously

[1] See Lander and McKay, Low Temperature Carbonization, p. 103.

shorten the time until combustion starts and will result in rapid ignition.

14 Returning again to the combustion process, the burning of the volatile matter produces water vapor and carbon dioxide, both of which will absorb and also radiate a considerable amount of radiant energy. The combustion of the volatile matter may be considered a step-by-step process where the hydrogen first unites with oxygen, freeing the carbon in an atomized state but highly heated by the burning hydrogen. Undoubtedly a certain amount of heat resulting from the combustion of the hydrogen is absorbed and dissipated as radiant heat by these free carbon particles, which are nearly pure black bodies and which give luminosity to the flame. Experiments have shown that luminous flames with incandescent particles of carbon floating in them radiate more heat than non-luminous flames. More radiant heat is emitted and may be absorbed from the flames of a pulverized-coal furnace than from those of an underfeed-stoker furnace due to this large volume of flame of greater luminosity. Much of the secondary combustion in the flame with stokers is due to carbon monoxide, which burns with a non-luminous flame and hence has less radiating power. The rapidity with which the free carbon particles in the powdered-coal furnace combine with oxygen depends very largely on the speed of mixing with heated air. This is also true of the coke which now forms the remainder of the original coal particle.

15 The speed of combustion of this coke and the consequent length of flame travel are dependent upon rapid mixing in the furnace. Unless the products of combustion of the volatile gases are swept away from the coke, it remains surrounded by inert gases. Oxygen cannot reach the particle until this gas envelope is removed farther along in the flame travel. The coke takes an interval of time to burn, and fresh supplies of oxygen must reach the coke throughout the whole combustion period. The extent of mixing necessary can be appreciated from the fact that the particle of coke must come in contact with many thousand times its own volume of air under furnace conditions, in order to burn completely. Violent mixing is thus an essential of rapid combustion and short flames. Turbulence of the mix has been found desirable for rapid ignition in solid-injection oil engines. Turbulence in the furnace seems to be an essential factor in the burning of powdered coal. Tests made on the Fuller tangential-jet furnace are said to bear out this opinion, and indicate that more rapid combustion with the development of more B.t.u. per cu. ft. than is now current practice may be obtained by violent turbulence in the furnace.

16 It is hardly necessary to point out that finely ground coal permits more thorough mixing and more rapid combustion. Hence the desirability of fine grinding.

17 The combustion just described cannot proceed until the

secondary air itself has been warmed up to ignition temperature, and this must be done by conduction and convection from the heated particles of carbon. This is a slow process. It will at once be evident that highly preheated secondary air is an accelerator of combustion and should therefore be employed.

18 Summing up the factors in the combustion process, it is desirable to use thoroughly dry and finely ground coal with highly preheated primary and secondary air. A state of violent agitation should be maintained inside the furnace to insure rapid contact of combustible and oxygen. Rapid combustion with consequent high furnace temperature and shorter flame travel than are present practice should result from the application of these principles.

Furnace Walls

19 Experience with furnaces as now designed indicates that more excess air is supplied at various ratings than is necessary for combustion, in order that the temperature may be kept within limits set by wall conditions. It would therefore appear that the desired more rapid ignition and combustion with resultant higher furnace temperature can not be permitted with certain wall constructions. Hence an analysis of furnace walls must be made to determine what construction will prove satisfactory.

20 The burning coke particle in the furnace throws out great quantities of radiant heat besides that emitted by the nascent carbon during the combustion of the volatile matter. The glowing ash particle that remains after the combustion of the coke is a gray or reddish body, and therefore absorbs and emits little radiant energy. In fact, it has very little heat to give up in any case. These facts account for the relatively cool condition of the ash below the water screen. The term " water screen " has been applied to those tubes which cross the furnace above the ashpit. They form part of the water circulation of the boiler and by absorbing radiant heat, cool the falling ash and the ashpit itself. Water vapor and carbon dioxide resulting from the combustion of the coal particle are at high temperature and, as noted before, emit radiant heat. Recent investigations[1] indicate that these radiations increase with depth of flame and in consequence may assume large proportions in the big furnaces usually employed under boilers.

21 Radiant heat is given off in all directions. In a pulverized-coal furnace the presence of such solid particles as coke, burning carbon, and incandescent ash particles are obstacles in the path of radiations from the interior portions of the flame and either absorb or reflect these radiations. It is therefore logical to assume that the furnace walls only affect the radiations of a relatively thin envelope of the furnace volume and that the interior of the furnace

[1] New Data on Heat Radiation, *Mechanical Engineering*, January, 1925, p. 41.

is affected but little, if any, by wall conditions. This idea is in conformity with certain researches by physicists and has been fully discussed elsewhere.[1] Such a deduction is highly significant, for it leads to the conclusion that wall temperatures cannot have much effect on furnace conditions as a whole. Hence it is possible to use water-cooled furnace walls.

22 Furnace walls are a vital part of the powdered-coal system and have a great effect on maintenance costs and boiler outage. Their construction must be greatly influenced by the character of the ash. The better-grade coals with high fusing ash are becoming exhausted in many coal regions, and during the life of the stations now being built much coal may be burned that will have higher ash and higher pyritic sulphur than the coal now used. Furnaces should be designed to meet such unusual conditions.

23 Experiences in the Middle West with low-grade coals indicate that air-cooled furnace walls are washed away rapidly by the chemical alloying properties of liquid ash containing a large percentage of iron oxide. Such liquid ash readily combines with the silica and alumina of the firebricks to form iron-silica-alumina compounds of the series well known in the cement industry. It does not appear to be possible to stop this alloying action as long as liquid-ash particles are thrown against white-hot firebrick walls. Air cooling of side walls only becomes effective when the bricks have washed down to thicknesses ranging from $\frac{3}{4}$ in. to 2 in. No neutral bricks have so far been tested that will stand up satisfactorily under furnace conditions and will not spall. These experiences would indicate that silica-alumina brick can only be used in air-cooled walls when high-fusing-point ash is present. When brick-lined furnaces are desired this wall may take the form of a water-tube-supported thin wall as in the Bettington boiler or of certain other forms now under development.

24 There is some question whether brick are needed at all in the furnace walls. More or less slag formation will always occur with low-fusing-point ash on any brick inside the furnace, and this will give trouble. Many engineers believe that brick walls are needed as reflecting surfaces to accelerate the ignition and combustion of the entering coal and that bare watercooled walls would absorb so much radiant heat that either long flame travel will be necessary or incomplete combustion will result. It has already been pointed out that only the outer envelope of the furnace flame will be affected in any case, and even then the net cooling effect will be minimized if the air and fuel are preheated and if violent turbulence exists. In any case the front walls may be of brick according to present practice in certain systems of firing. Care must be taken in designing front walls that the brickwork properly shields all supporting steel from radiant heat. Air-cooled brick

[1] The Influence of Radiant Heat on Furnace Design, *Power*, May 29, 1923, p. 851.

walls or brick-faced water-cooled walls may be necessary to give reverberatory action in furnaces burning anthracite, coke dust, or low-volatile coal.

25 Water-cooled furnaces should logically permit higher boiler capacities to be carried with a given total heating surface for the boiler unit and a given furnace volume. The radiant heat from the flames is absorbed by the metal walls and transferred directly to the water. The rate of heat absorption by radiant heat is about eight to ten times the average rate by convection. Data now at hand indicate that the rate of heat transfer by radiant heat to metal side walls may reach as high as 75,000 B.t.u. per sq. ft. per hr. In the case of brick-lined furnaces the radiant heat is reflected by the brick surfaces, and while some of this reaches the boiler tubes, much is absorbed by the gases and raises their temperature. This heat is carried up into the farther passes of the boiler where it is given up to the tubes by convection at a relatively lower transfer rate than by radiation. Hence the total heating surface must have lower capacity with brick-lined furnaces than in the case of the metal-lined furnace.

26 Completely water-cooled walls, both rear and side, seem to be fully warranted. The ash will have no washing or cutting effect on these, and probably little trouble will be experienced with slag. Since radiant heat is the most effective of the three methods of heat transfer, this construction should tend to produce still higher boiler efficiency, and as this furnace wall surface is highly effective as a heat absorber, less surface need be put in the upper sections of the boiler. The resulting reduction of boiler height would require a lower boiler room and hence a less costly plant.

27 There is one factor in the construction of metal side walls that does not seem to have been given any attention. It is a well-known fact that gray surfaces are poor absorbers of radiant heat and therefore are largely reflectors. If the outer surfaces of the metal tubes could be made of a light-gray material instead of black iron, the absorption of radiant heat would be decreased and the cooling effect on the borders of the furnace flame would be diminished. Calorized tubes might be tried for this service. This construction might be used where coke or anthracite are burned and less cooling by the side walls is desired.

28 There appear to be no insuperable difficulties in the construction of a satisfactory water-cooled furnace lining. Such construction would permit the highest attainable furnace temperatures. As stated before, minimum excess air is not employed in the brick-lined powdered-coal furnaces, for the walls will not stand the high furnace temperatures that would be developed. This limitation would be removed with metal walls and boiler efficiency would be thereby increased. A portion of one of the walls might be occupied with a radiant-energy superheater which operates either alone or in series with a convection superheater.

Air Preheaters

29 The ideal power plant with powdered coal would be one where the feedwater is heated solely by bled steam and the heat in the chimney gases recovered by air preheaters. Tests at Colfax indicate that there are gains to be made by preheating air over and above that due to the increase of air temperature. These gains appear to be due to the more efficient transfer of heat by radiant energy as a result of higher furnace temperatures. Furthermore it will be apparent that this air should be preheated to as high a temperature as can be secured. While air preheaters are still in the development stage, satisfactory progress has been made already and improved designs can be expected.

30 One feature of air preheaters frequently overlooked is their ability to provide an adequate supply of outside air for combustion in winter time through ducts for this purpose. Boiler-room operators object to working in a cold, drafty building, and very often an adequate supply of air is lacking during the winter months, for all doors, windows, and other openings to the outside are kept closed. The air preheater should be credited with some of the advantages of such positive air supply.

31 Attempts have been made to preheat air by bled steam. The use of heaters similar to those now in use for cooling generator air have been considered for this service. There are possibilities of slight gains in plant economy by this process. Insufficient data are available on heat transfer in such heaters at high temperatures, and performance can not therefore be definitely predicted. The scheme warrants further study.

Coal Driers

32 Coal drying is desirable as previously pointed out, aside from the fact that coal transports better when thoroughly dried and that seasonal difficulties from frost and snow are eliminated. Coal may be dried by either of three methods. It may be furnished from some outside source with sufficient heat to raise its temperature to the point where its water content will vaporize, force itself out to the surface of the lumps, and separate out as steam. This process is used in certain rotary driers. A second process is to pass heated gases through the coal and evaporate its moisture content by humidifying the gas. The controlling factor in this method is the wet-bulb temperature to which the coal is heated while evaporation takes place, and not the dewpoint as some have stated. This plan is used in the waste-heat driers and steam driers now in use. The third method is a combination of the two previous methods and is employed in the standard rotary drier. The outer shell is heated by the fire and by the hot gases which afterward pass through the driers, evaporating moisture according to the second method.

33 The first method, while commercially practicable, is not as efficient as the other two. It is used generally for drying substances that cannot be allowed to come in contact with flue gases. It is not recommended for pulverized-coal plants.

34 The third method, employing the rotary drier, was formerly used in all powdered-coal plants. It is an effective means for drying coal and can reduce the moisture content of the coal to a low figure. The objections to its use are the amount of space required to house it, the fuel required for drying, and the dust nuisance from its waste gases. Thoroughly dry coal can be pulverized in screen mills with a lesser expenditure of power than in air-separation mills. This fact coupled with the advantage to be derived in the furnace from thoroughly dry coal may more than offset the other disadvantages of the rotary drier. This system has been adopted in some recent installations whose performance will be followed with interest.

35 The second system has been installed in many plants. Its operation is not so well understood as those just discussed and its limitations must be fully considered. The waste-heat drier uses a portion of the flue gases from the boiler. These are drawn by a forced- or induced-draft fan through a layer of coal usually less than a foot thick. The evaporation of moisture tends to saturate the gases during their passage through the coal.

36 Hot spots were frequent in some of the first of these driers used on high-sulphur coal. At first it was thought undesirable to admit high-temperature gas to these driers and the gas was therefore diluted with air, which provided oxygen to start spontaneous combustion of coal dust on ledges and elsewhere in the drier. The practice of diluting with air was abandoned when it was realized that the coal remains at wet-bulb temperature while evaporation takes place, and thus practically throughout its whole passage through the drier. Flue gases as hot as available are now used. The oxygen content in these gases is too low to start fires and the troubles of spontaneous combustion have disappeared.

37 Certain difficulties still remain. Some coals readily give up their surface moisture but, either from lack of capillarity or due to the presence of combined water, give up little of their interior moisture. When such coal reaches the pulverizers the work of attrition in crushing to small size liberates the water, so that these mills and their cyclones have to be well vented to remove this moisture. These vents discharge some coal dust, which adds to the dust nuisance.

38 However, for low-moisture coals, particularly where this moisture is largely on the surface, such driers are quite satisfactory. Some development work must still be done on these driers to insure more thorough drying with a smaller quantity of flue gases. Two passes for the gas have been suggested. The first pass of the hot gas would be through the cold entering coal and the second

pass lower down through the coal about to leave the drier. Neither coal nor gas would then fall below wet-bulb temperature. This would require less gas than the single pass and would insure more thorough drying. The coal stream could be made thinner than present practice.

39 Steam driers are used in certain cases where waste gases are not available and where low-pressure or bled steam can be had. Air is drawn through a set of nested heating grids which form the chute down which the coal passes. Coal is a poor heat conductor, hence little will be heated above the wet-bulb temperature of the entering air. This wet-bulb temperature will be much lower than that of the flue gases in the waste-heat drier, and although oxygen is present, there is little danger of fires at this low temperature.

Ash

40 Certain coals have such high-fusing-point ash that no difficulties are experienced from this source at any ratings that may be carried on the furnace. Hence these plants can operate with hollow-walled brick furnaces and simple ash-handling equipment. However, as already pointed out, many stations may be compelled during their life to use lower-grade fuels often containing a high percentage of pyritic sulphur, and this must be considered in the design of the plant. Such ash slags and washes badly on brick walls. However, it will probably adhere little to water-cooled walls, and while occasional clinkers may form on the water screens and on the lower boiler tubes, these can generally be easily removed.

41 A portion of the low-fusing ash forms sharp, sand-like particles which fall to the bottom of the furnace together with some ash dust. The removal and transport of this ash present some problems. Its abrasive properties have proven destructive to ducts where water flushing has been employed. It would be difficult to remove this material in a dry state on account of its high temperature and dust. Steam-jet ash transport is uneconomical of steam and requires a large make-up for the station. Some new and better scheme of handling the ash from the furnace bottom must still be devised.

42 Some of the finer dust particles that pass through the boiler may be caught by cindervane fans. The very fine particles have been allowed to pass up the chimney to be discharged at the top with the flue gases.

43 Engineers have already done much to lessen the dust nuisances of all our large cities, but more can still be done. It should therefore be a public duty to reduce as far as possible the discharge of dust and smoke from chimneys of stoker plants and of those using pulverized coal. Such dust floating in the air lessens the amount of sunlight reaching the surrounding earth, increases fog conditions, increases the cost of keeping cities clean, increases

disease, and lessens the health and vitality of those who breathe the air it contaminates. There are thus good social reasons for minimizing the discharge of dirt from chimneys of all kinds, including locomotives.

44 Several methods have been proposed to catch this ash in new super-power plants. Washers lower the gas temperature and lessen the effectiveness of the chimney as a draft producer. Their success in cleaning air, however, warrants further experimenting on flue gases. The Cottrell electric precipitation system works satisfactorily on cement dust and is said to be efficacious as a remover of ash from the flue gas at the Trenton Channel Plant of the Detroit Edison Company. Other schemes have also been proposed but have not been fully demonstrated. This whole question of dust nuisances has not been given as full attention as it deserves.

45 The fine dust from chimneys should find some commercial use in the industries either as a filler for paints, fertilizer, or other products, or as a base for some building material. Such uses will undoubtedly be found in the near future, for much money has been spent on development work.

Conclusions

46 This analysis of boiler furnaces for pulverized fuel indicates that for highest efficiency and maximum capacity:

(*a*) The coal should be thoroughly dry, and be finely ground and preheated before admission to the furnace.

(*b*) The air entering the furnace should be highly preheated, and this can be done best by flue gases, although there are possibilities in the use of bled steam.

(*c*) Turbulence should be maintained in the furnace to insure rapid combustion and high efficiency.

(*d*) The furnace walls should be water-cooled, preferably with all-metal walls. This construction insures low boiler outage, low maintenance, and high efficiency of heat transfer. Water-cooled walls with brick facing may be desirable with certain fuels.

(*e*) Improved methods should be developed to handle ash from any coals, and the discharge of dust and smoke from the top of all chimneys should be reduced.

(*f*) To secure increased furnace capacity and at the same time secure higher boiler efficiency, more B.t.u. per cu. ft. of furnace volume must be developed. This is possible by employing the factors noted above.

DISCUSSION

C. G. SPENCER.[1] The metal-lined furnace for pulverized coal which was put in service about eight weeks ago at Cahokia has given good results. The boiler has not been taken off the line during this time because of furnace trouble. The performance confirms Professor Christie's analysis, showing a decrease in furnace temperature, no slagging when operating at a higher CO_2 than can safely be carried in furnaces not metal-lined, and no fouling of the metal lining.

Fin tubes cover the side and rear walls. Above the ashpit is a screen of plain tubes spaced to allow ash to drop through. The front wall is of firebrick, air-cooled, with ports to admit secondary combustion air.

Since this is the first pulverized-fuel furnace to go into operation where metal lining has been carried to this extent, a few general observation notes may be of interest.

The boiler was first operated with an interdeck convection superheater of the same surface as installed throughout the plant. This surface was insufficient because of the depressed furnace temperature, and another row of superheater tubes was added, thus increasing the surface by about one-third.

The draft drop through the boiler was noticeably less because of the reduced gas volume. This is the direct result of the higher CO_2 permitted by the metal lining and allows of operating the boiler at a higher rating with the same stack draft.

The ash from this furnace is granular or spongy. It is free from the hard vitrified clinker which is characteristic of Illinois coals when burned in a higher-temperature furnace.

The circulation gives every indication of being satisfactory. The horizontal headers outside the furnace into which the vertical fin tubes are expanded are connected by outside vertical tubes which provide a closed circulating system independent of the boiler. Because of this, the downcomer from the water space in the boiler drum becomes a feed line to make up for the water evaporated in the fin tubes, and the uptake becomes a steam delivery line.

W. J. WOHLENBERG.[2] Professor Christie has outlined the fundamentals involved in the design of pulverized-fuel furnaces in a clear and concise manner. The material as a whole appears to be exceptionally well presented. One or two statements may be commented upon for further clarification.

In Par. 5 we find the statement, "On the other hand, many insulating materials, such as firebrick, reflect the radiant-energy

[1] Engineer, McClellan & Junkersfeld, Inc., New York, N. Y. Mem. A.S.M.E.

[2] Associate-Professor of Mechanical Engineering, Sheffield Scientific School, Yale University, New Haven, Conn. Assoc-Mem. A.S.M.E.

waves from their surfaces and absorb little heat." It is true that refractory materials absorb little heat, but in general this is due to the fact that they are very poor conductors rather than good reflectors of heat. The energy emitted from such surfaces is therefore, for the most part, re-radiated in all directions rather than reflected in one direction. Carbon is another example of a material which is both a poor reflector and conductor of heat, while at the same time it is very nearly a perfect absorber of energy as radiant heat. These questions are discussed in more detail in the paper[1] by Mr. Donald G. Morrow and the writer.

In Par. 21 the author assumes that the furnace walls affect only the radiations of a relatively thin envelope of the furnace volume. In view of an analysis included in the paper[1] already mentioned, it appears that this would be the case only if the average coal particles were very much smaller than they actually are, as now burned in the furnace. It is true that the temperature in the flame interior is higher than at the outer boundary. This is probably due largely to a higher rate of energy liberation in such interior spaces, the shielding of radiations under present furnace conditions, although not negligible, being by no means complete with respect to the interior portions of the flame. If we consider a reduction in particle size until the shielding is complete, due to the resultant large increase in the number of particles, there would still be radiations from the shielded to the partially shielded and completely free particles, which in turn would radiate to the walls. Such conditions have been designated as "series radiation" in the paper mentioned. Even under such conditions the central portion of the flame would be somewhat affected by radiations to the furnace walls unless the particles should become indefinitely small and hence infinite in number.

At the other limit a single free carbon particle might be considered as radiating to the furnace walls. The temperature attained by the particle would then be determined by the rate at which energy is liberated at its surface. This is determined, of course, by the rate at which oxygen is brought in contact with its surface, which in turn depends on the relative velocity between the particle and the oxygen or air atmosphere within which it is moving. It appears, then, that under the proper conditions any desired flame temperature might be realized, even in a furnace wholly surrounded by cold surface, and that, therefore, we may not assume a condition of complete shielding merely on the evidence of higher interior flame temperature alone.

E. G. Bailey.[2] The author has made a very clear statement of the nature of combustion and methods of heat transfer, and then,

[1] Radiation in the Pulverized-Fuel Furnace, see page 127.

[2] President, Bailey Meter Company, Cleveland, Ohio. Mem. A.S.M.E.

in the writer's opinion, reached a wrong conclusion in Par. 21 where he recommends cold furnace walls.

The crucial point in getting the best efficiency from pulverized coal is to completely burn the floating particles of carbon in the "thin" part of the flame. With the furnaces of large volume in general use today, the writer believes that about 90 per cent of the combustion takes place in about 10 per cent of the furnace volume, while the other 90 per cent of volume is required to complete the last 10 per cent of the combustion. It is the last 1 per cent or fraction thereof in which we are greatly interested. In that part of the furnace where the flame is thin and transparent the laws of radiation must be carefully considered and the correct wall temperatures maintained. Otherwise these particles of burning carbon will radiate their heat to the cold walls and drop below the ignition temperature, even while floating in gas many degrees hotter.

The particles in this zone burn slowly because they are in an atmosphere of about 98 per cent inert gases where they are endeavoring to contact with the remaining 2 per cent or so of oxygen. Their temperature cannot be maintained above the temperature required for complete combustion when they can "see" cold walls. The flame is certainly thin enough for the particles to "see" the walls when one can look through the flame across a furnace 20 or 30 ft. wide and distinguish every detail of the opposite wall.

If cold walls have no ill effect upon combustion, then why so much smoke when starting a fire in a cold furnace? And the smoke persists until the walls are hot. Just how hot they must be maintained for the complete combustion of the last 1 per cent of floating carbon remains to be seen, but it probably is considerably above the temperature of bare water-cooled walls.

With the advent of turbulent burners, hot walls will still be necessary. While the furnace volumes will be considerably reduced, so will the luminosity of the flame and there will always remain the problem of burning the last 1 per cent of carbon floating in a transparent atmosphere where the percentage of oxygen is so low that heat cannot possibly be developed as fast as it will radiate to cold surroundings.

H. W. Brooks.[1] Professor Christie in this paper and Professor Wohlenberg in his[2] both stress the importance of rapid mixing of gases in the furnace to secure rapid and complete combustion. On this the speed of combustion and length of flame travel, and hence the combustion volume, are dependent. Few engineers understand the extent of this mixing necessary.

[1] Consulting Engineer, Fuller-Lehigh Company, Fullerton, Pa. Mem. A.S.M.E.

[2] Radiation in the Pulverized-Fuel Furnace, see p. 127.

To visualize the action that takes place in the burning of a particle of coal just passing a 200-mesh screen, consider the particle to be enlarged to the size of the airship *Los Angeles* (650 ft. long, total volume 2,700,000 cu. ft.). The comparative size of the oxygen atom would be somewhat less than an 0.08-in. cube, or, let us say, about the size of a gnat.

Picture then the *Los Angeles*-sized coal particle moving through the air either horizontally at a rate of somewhat less than a mile a minute, or vertically at somewhat more than a mile a minute, carrying entrained with it some twelve thousand billion gnat-sized oxygen atoms (30 per cent excess air assumed). All these gnat-sized oxygen atoms, in addition to their lineal speed, are vibrating in fixed orbits at velocities many times the speed of a bullet, and measured in miles per second. Complete combustion necessitates the *Los Angeles*-sized coal particle disintegrating into some nine thousand, three hundred billion carbon atoms and combining with a corresponding number of gnat-sized oxygen atoms.

In stoker firing, disintegration of coal is mechanically aided but little. In the use of pulverized coal, natural processes are assisted somewhat by reducing the size before firing, but only in part have the natural processes of diffusion and atomic attraction bringing the carbon atom and the oxygen atom together been aided. Like Professors Christie and Wohlenberg, many writers on combustion in the past have stressed the importance of turbulent flow, agitation, and mixing to aid the natural diffusivity, but a practical means of accomplishing this mixing has heretofore remained undiscovered.

Several years ago it occurred to one of the engineers of the Fuller-Lehigh Company that one of the most intense manifestations in nature of turbulent flow and agitation of gases and particles in suspension was in the tornado, where it had been repeatedly demonstrated that materials of considerable tensile strength had actually been torn apart, disintegrated, and reduced to fine pieces and sometimes to powder by the intense centrifugal action of the air. Experiments were started at Fullerton on a small furnace 18 in. square by 3 ft. deep in which the jets were placed to throw the flame tangent to the tornado of the fire within the furnace, the flame of the first jet being deflected before it reached the refractory walls by the impingement with equal velocity at right angles of the flame from the second jet, the third jet again changing the direction 90 deg., and the fourth jet completing the tornado within the pot.

In order to give a clearer idea of the placing of these jets, Fig. 1 shows in plan the application both to the circular- and the square-pot furnace.

It was well known that the external walls of the tornado of nature had been observed to be rather sharply defined from the surrounding air. Thus it was decided to adapt this principle by

placing the pulverized-fuel jets in such relation to the refractory walls that there was little, if any, impingement of the fuel tornado on the furnace walls. As was anticipated, therefore, the refractory

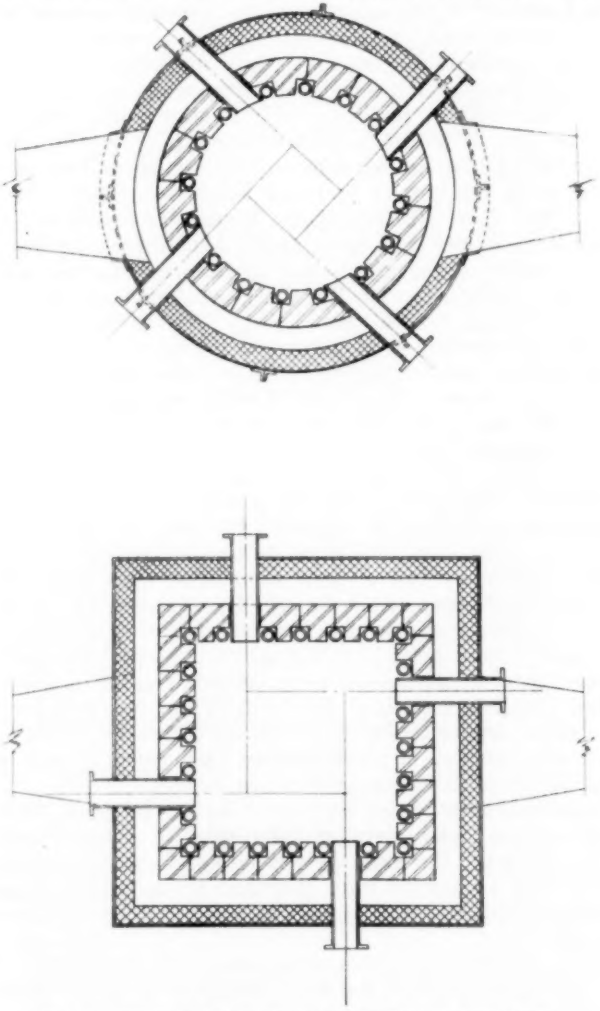

FIG. 1 TYPES OF WELL FURNACE FOR PULVERIZED FUEL

walls showed little damage due to erosion, and thermocouple temperatures taken on the inner refractory surface of the furnace showed that the inner face of the brick never exceeded 2700-2900 deg. fahr

It had long been recognized that the transmission of sensible heat from the hot gases from the furnace through the tubes of the boilers to the water on the other side was limited by the existence at the boundary surface of a "dead film," generally assumed to be about 1/40 in. thick, of relatively cool gas in which convection currents apparently did not take place, and which owing to its low conductivity offered a high resistance to the passage of heat (other than radiant) through it. This "dead film" is the main and most potent obstacle to rapid heat transmission in the boiler, and it follows that its complete or even partial destruction will greatly assist the flow of heat and hence the efficiency of the boiler. Prof. J. T. Nicolson of the Manchester School of Technology demonstrated a formula proving that the heat transfer was a direct function of the velocity of molecular travel of gases passing the tubes. According to Dr. Wilhelm Nusselt, this transfer varied as the 0.786 power of the gas velocity. Thus it was anticipated and tests subsequently proved that with the higher scouring action of the swirling gases in meeting the boiler tubes that there would not only be a higher rate of heat transfer and higher boiler efficiency, but also a cleansing action which would keep the tubes clear of deposited ash.

Early during the experiments the nature of the ignition and combustion taking place in the well proved quite unlike anything experienced pulverized-fuel engineers had hitherto seen in powdered-coal combustion. The flame itself resembled that of a blow torch, virtually a "ball of fire," combustion apparently taking place in a limited zone within the well. By regulating air admission and air pressure it was possible to move the hottest zone into the well itself or to a point just in front of the well. Appreciating further the gains due to the more efficient transfer of heat by radiant energy as a result of this "ball of fire," the inventor had still further reason to expect efficiencies higher than had been before demonstrated in pulverized-fuel combustion, as well as a flatter overall performance curve at high ratings.

These experiments were kept secret for several years, and although several larger operators volunteered to install a furnace of the new type, the company felt it best to withhold a commercial-scale installation until they had finally and thoroughly convinced themselves that no practical operating difficulties would intervene.

Finally on February 5, 1924, an agreement was entered into with the United Electric Light and Power Company of New York City by which a commercial-scale experimental installation of the new furnace would be made at their Sherman Creek Station alongside five other boilers fired by the older pulverized-fuel-firing methods being exploited by various manufacturers. By thus conducting six simultaneous tests on six different systems under similar conditions — and entirely with their own personnel — the new furnace

construction would not only be subjected to a large-scale test, but at the same time could be directly compared with five competitive systems. Today it is understood these tests have been completed. It had been our hope to be able to present them at

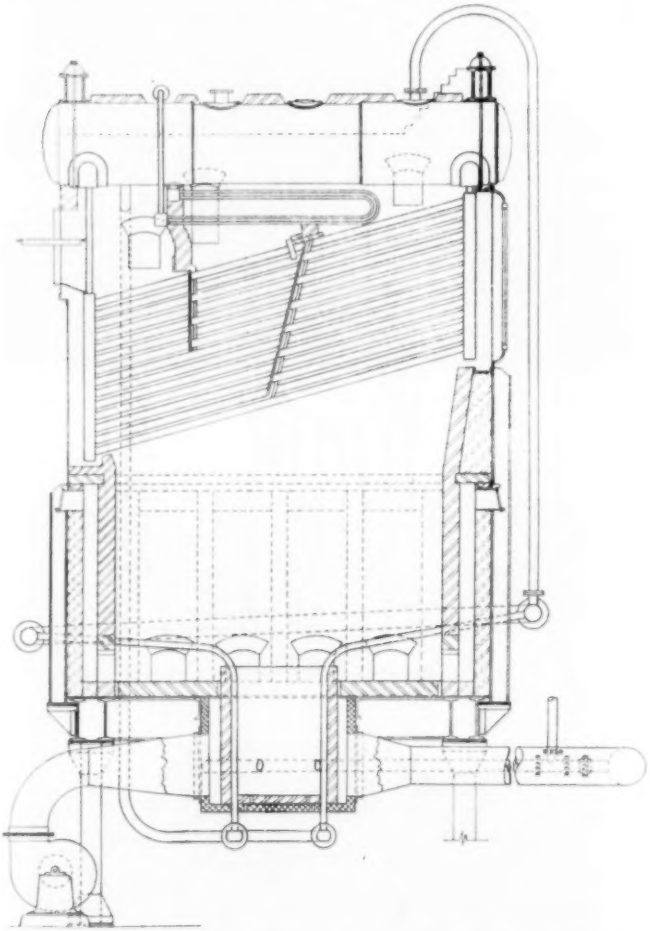

FIG. 2 WELL FURNACE APPLIED TO WATER-TUBE BOILER

this meeting. This having not proved convenient to the authorities of the United Light and Power Company, we shall look forward to the privilege of their presentation at some later meeting. We are permitted to state, however, that the efficiencies have actually proved from 3 to 5 per cent higher than with any other method

of firing, and the curve of efficiencies at various rates of driving has been much flatter than anything heretofore demonstrated.

Unfortunately, there is not available a design of the furnace used at Sherman Creek. In its place Fig. 2 is shown indicating one

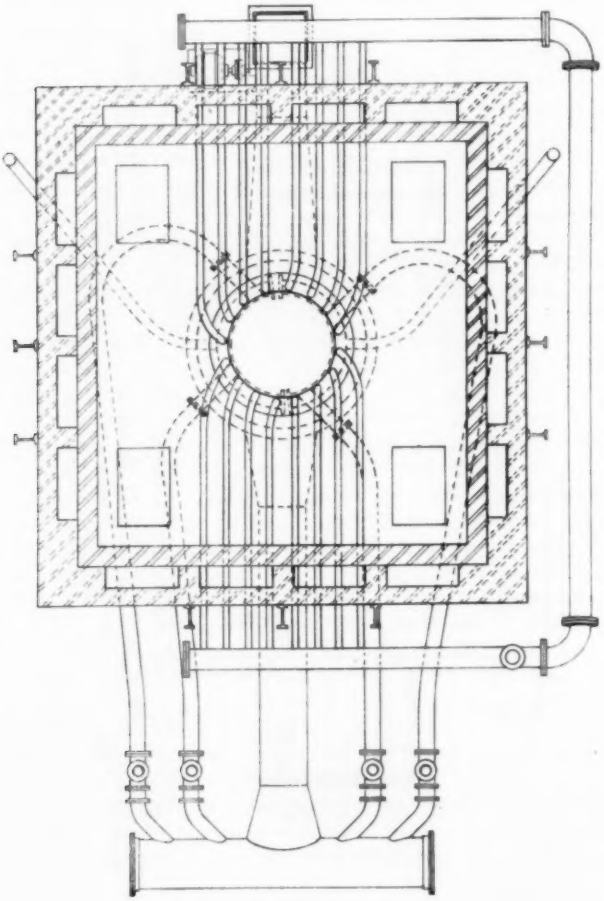

FIG. 3 WATER-COOLED WELL

of the more recent applications of this furnace. It will be noted that above the well there is provided a dispersion chamber in which the hot gases after burning are allowed to expand before reaching the boiler tubes, thus reducing the tornado effect to a value which can cause no possible erosive effect on the boiler tubes. The theory

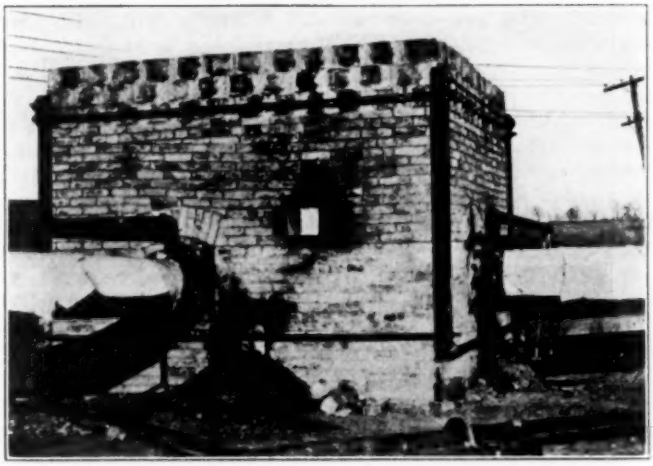

FIG. 4 WELL FURNACE, 8-FT. CUBE, FOR EXPERIMENTAL WORK

FIG. 5 WELL FURNACE JUST PRIOR TO IGNITION

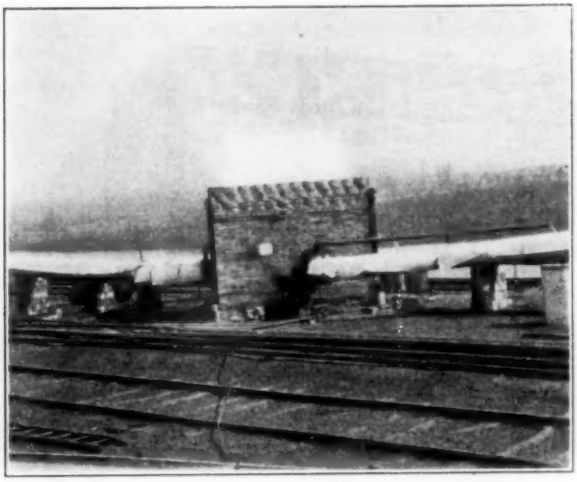

FIG. 6 WELL FURNACE JUST AFTER IGNITION

FIG. 7 8-FT. WELL FURNACE BURNING THE COAL EQUIVALENT OF A 1500-HP. BOILER AT 632 PER CENT OF RATING

of this chamber was to allow just sufficient velocity to scour off the "dead film" from the boiler tubes without going any further in attacking the tubes themselves. It is not possible to state at this time the minimum size to which this chamber may eventually be reduced. It has definitely been established, however, that the combined volume of well and dispersion chamber can be made substantially less than the combustion space provided in the old-fashioned stoker setting. Knowing that we can convert present stoker-fired settings by this means, and attain the highest boiler ratings desired, we have at this time no occasion to go further until the application is extended to other services, such as transportation and marine boilers.

Fig. 3 shows a plan of the water-cooled well as actually applied on the later designs.

Simultaneously with the tests at New York research has been continued at Fullerton on a furnace shown in Fig. 4, consisting of an 8-ft. cube. Fig. 5 shows the furnace being ignited, and Fig. 6 the furnace immediately after ignition. Fig. 7 shows the very remarkable phenomenon of the coal equivalent of a 1500-hp. boiler at 632 per cent of rating being burned in the volume of an 8-ft. cube.

It is thus felt that today the "well type" furnace is a demonstrated success and that for the first time in the history of pulverized coal the problem of turbulent flow and hence the problem of combustion volume has been solved. Not only may pulverized coal be efficiently burned in combustion volumes as small as those previously employed for stokers, but actually smaller furnaces may be installed where necessary. For the first time, completely successful application of pulverized coal to transportation and marine service becomes possible, as does also the conversion of present stoker-fired furnaces to the advantages of pulverized coal without the necessity in many cases of more than slightly modifying the present furnace construction. Thus many operators who have heretofore found the application of pulverized fuel impracticable on account of high cost of furnace reconstruction have now made available to them the increased efficiency, economy, and flexibility of control of pulverized-fuel firing.

C. W. E. CLARKE.[1] If, as stated in Par. 21, the heat radiated to the furnace walls has little effect on the furnace temperature as a whole, there will be no material change in the quantity of heat to be absorbed in the upper parts of the boiler.

The total surface in the boiler, measured in terms of its effectiveness, that is, considering surface exposed to radiant heat as more effective than that exposed only to "convected" heat, must

[1] Consulting Engineer, Dwight P. Robinson & Co., New York, N. Y. Mem. A.S.M.E.

be the same for a given output whether the walls be air- or water-cooled. This surface must be properly distributed between the two classes of work just mentioned. If the gases entering the first pass are at the same temperature for both air-cooled and water-cooled side walls as is implied in Par. 21, then the amount of surface in the upper parts must be the same per pound of gas passing through the boiler. However, it is evident that if a very considerable amount of heat be absorbed by surface in the furnace walls the furnace temperature must be materially reduced, or else there must be more heat produced in the furnace to offset the radiant-heat absorption. The temperature of the gases in the first pass will be reduced with water-cooled walls, but due to this lower temperature the mean heat transfer per unit of area in the boiler proper will be reduced. It is very probable that to get the same temperature of exit, nearly the same length of gas travel in terms of surface passed over will have to be provided.

The tests made on boilers in the Colfax Station operating with preheated air showed that though the furnace temperature was measurably increased the exit gas temperature was reduced, an indication, the writer believes, that at least with temperatures of the order of 2800 to 3200 deg. fahr., higher temperatures result in more efficient working of the boiler furnaces. It is not probable that the question of boiler height can be answered by deduction. As indicated, the distribution of work through the boiler is materially changed by the introduction of considerable surface exposed to radiant heat. The efficiency of these radiant-heat surfaces will be materially higher, but the efficiency of the "convection" surfaces will be reduced perhaps sufficiently to offset it.

The whole matter may be summed up by a consideration of the fundamental fact that the relative efficiencies of two boilers may be measured by the temperature of their exit gases, assuming for comparison the same generated temperature in the furnace. By "generated temperature" is meant the temperature of the burning material. There is no question that if, as now appears to be the fact, the use of water-cooled side walls does not interfere with complete combustion, the average efficiency of the boiler surfaces will be increased, resulting in less total surface per unit of output. The writer seriously doubts, however, if any material decrease in boiler height (length of gas travel in terms of surface passed over) can be made without increasing the exit-gas temperature and hence lowering the efficiency.

In Par. 27 the color of the water-cooled surfaces is suggested as a factor influencing the efficiency of the surfaces. While the effect of color on heat absorption is a well-established fact, any attempt to control the color of metal surfaces exposed to furnace conditions must be fraught with extreme difficulty. These surfaces take their color from particles of ash and dust that adhere to them, and it is doubtful if this factor is controllable.

We have not found it necessary to dry coal for pulverizing in the Pittsburgh district where the moisture content is of the order of 4.5 per cent. However, in the operation of a 15-ton mill it was found that the introduction of preheated air to the mill resulted in a considerable improvement in its operation and in the feeder operation. When operating with cold air, the moisture from the coal collected in the mill and mixed with the coal to form a semi-

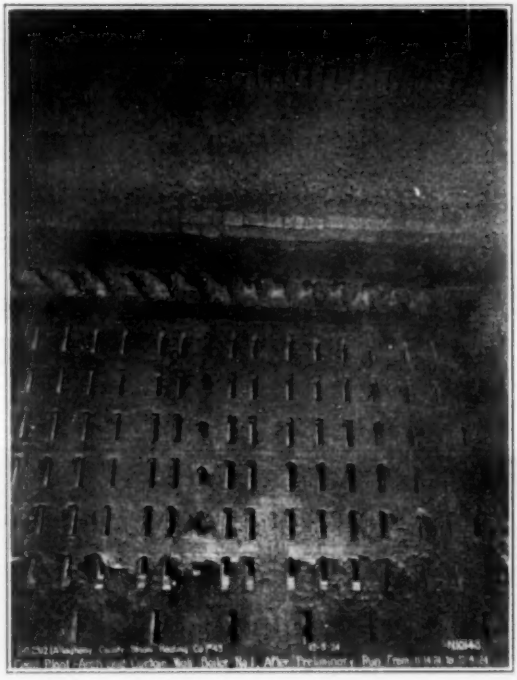

FIG. 9 ALLEGHENY COUNTY STEAM HEATING COMPANY, CECIL PLANT. LOWER FRONT WALL, BOILER NO. 1, AFTER PRELIMINARY RUN OF 480 HOURS

liquid soup, interfering considerably with proper operation. The heated air entirely eliminated this difficulty. When operating with cold air the coal had a tendency to pack in the feeders, and this also was eliminated when heated air was used.

The use of heated air in the mills has been objected to on the ground that it increases the loss through the vents. However, in using practically any type of drier there is an appreciable loss of dust from the drying coal and there is probably little difference in the loss by the two methods.

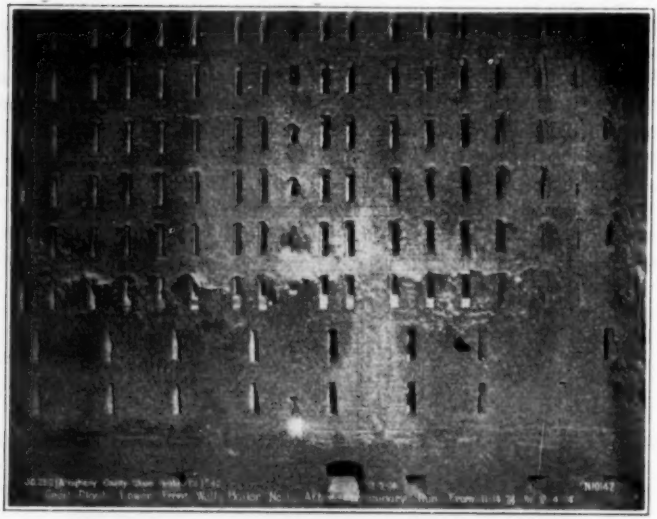

FIG. 8 ALLEGHENY COUNTY STEAM HEATING COMPANY, CECIL PLANT. ARCH AND CURTAIN WALL, BOILER NO. 1, AFTER PRELIMINARY RUN OF 480 HOURS

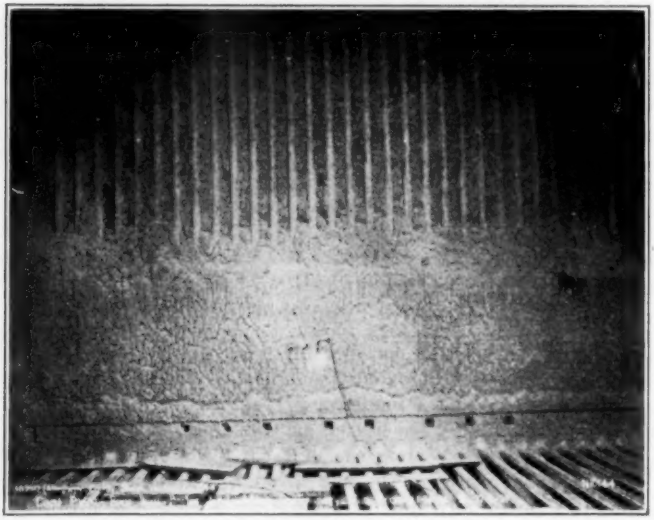

FIG. 10 ALLEGHENY COUNTY STEAM HEATING COMPANY, CECIL PLANT. REAR WALL, BOILER NO. 1, AFTER PRELIMINARY RUN OF 480 HOURS

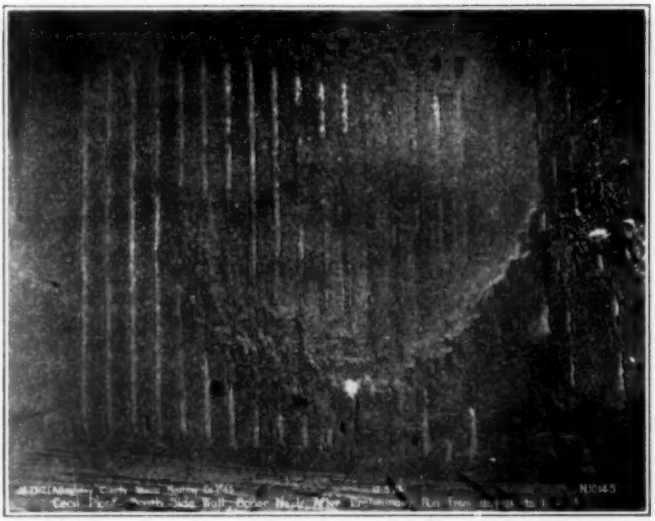

FIG. 11 ALLEGHENY COUNTY STEAM HEATING COMPANY, CECIL PLANT. SOUTH SIDE WALL, BOILER NO. 1, AFTER PRELIMINARY RUN OF 480 HOURS

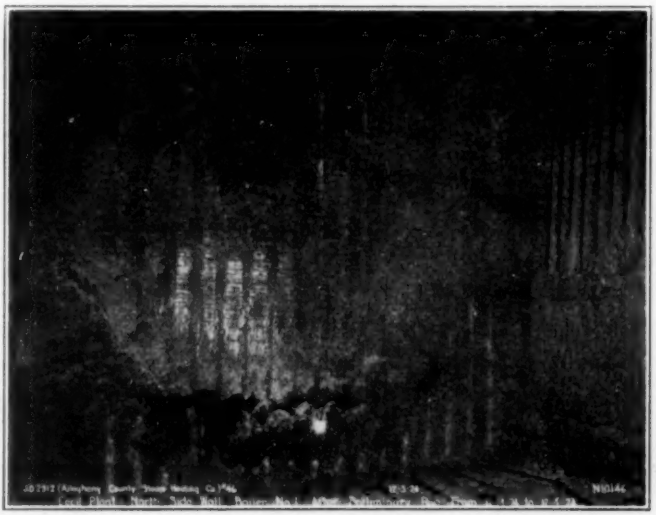

FIG. 12 ALLEGHENY COUNTY STEAM HEATING COMPANY, CECIL PLANT. NORTH SIDE WALL, BOILER NO. 1, AFTER PRELIMINARY RUN OF 480 HOURS

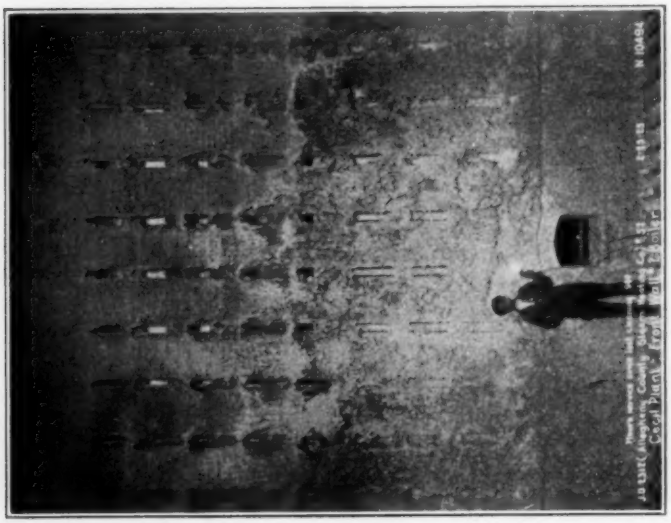

FIG. 13 ALLEGHENY COUNTY STEAM HEATING COMPANY, CECIL PLANT. LOWER FRONT WALL. BOILER NO. 2, AFTER 900 HOURS' OPERATION

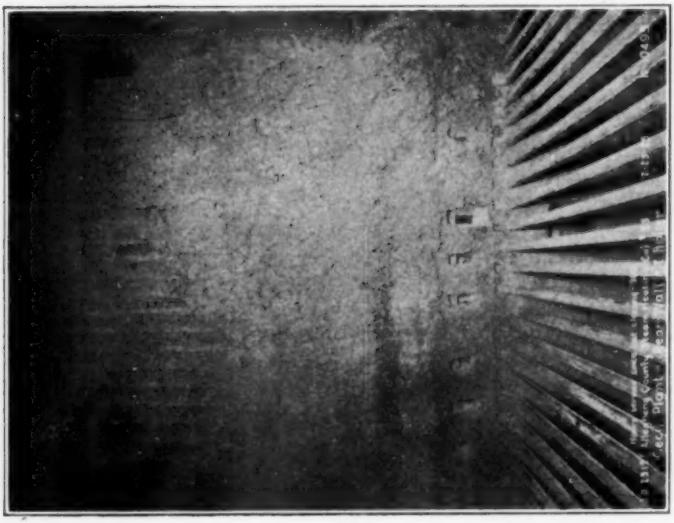

FIG. 14 ALLEGHENY COUNTY STEAM HEATING COMPANY, CECIL PLANT. REAR WALL. BOILER NO. 2, AFTER 900 HOURS' OPERATION

FIG. 15 ALLEGHENY COUNTY STEAM HEATING COMPANY, CECIL PLANT. SOUTH SIDE WALL, BOILER NO. 2, AFTER 900 HOURS' OPERATION

FIG. 16 ALLEGHENY COUNTY STEAM HEATING COMPANY, CECIL PLANT. NORTH SIDE WALL, BOILER NO. 2, AFTER 900 HOURS' OPERATION

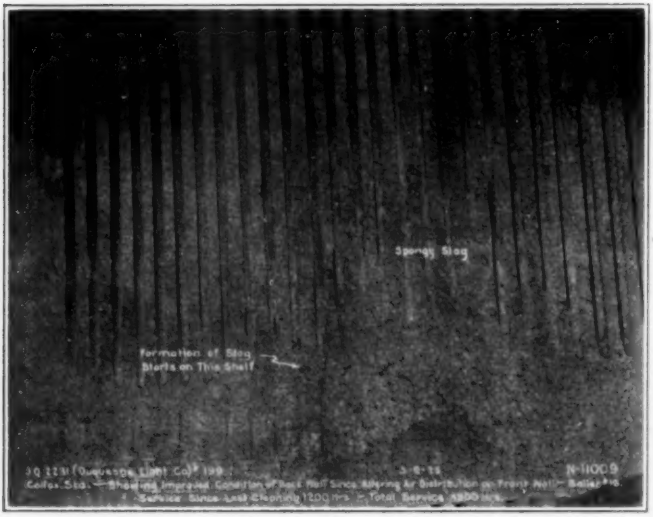

FIG. 17. DUQUESNE LIGHT COMPANY, COLFAX STATION. REAR WALL, BOILER NO. 16, AFTER CHANGE IN AIR DISTRIBUTION.
Service hours since last cleaned, 1200.

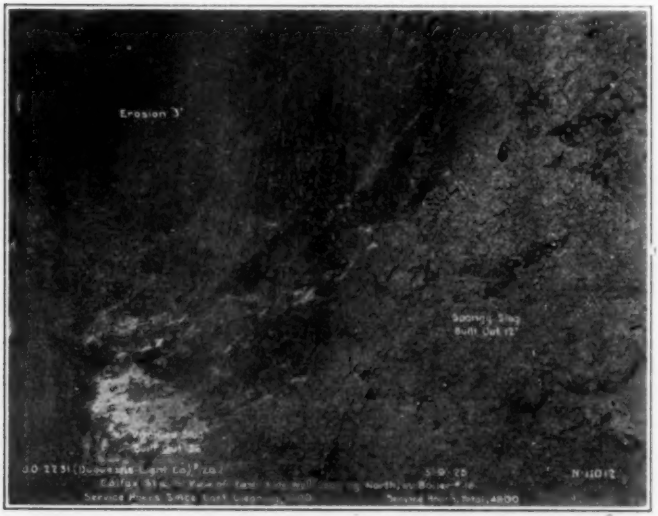

FIG. 18 DUQUESNE LIGHT COMPANY, COLFAX STATION. SIDE WALL, BOILER NO. 16, SHOWING EROSION ABOUT 3 IN. DEEP NEAR FRONT WALL AND SLAG BUILT OUT ABOUT 12 IN. BELOW ERODED AREA
Service hours since last cleaned, 1200.

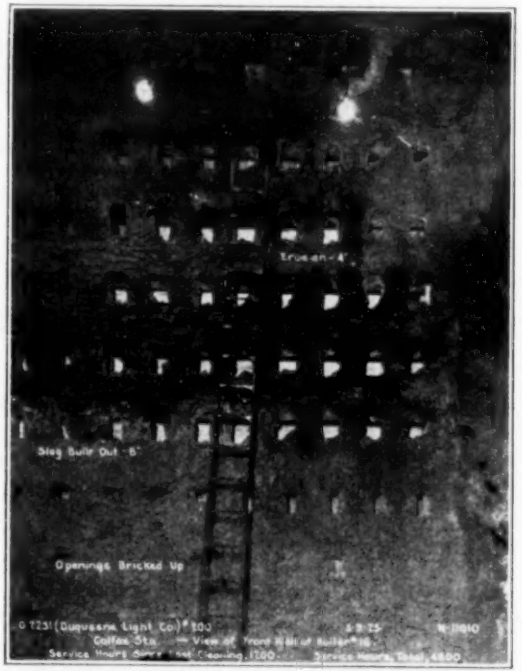

FIG. 19 DUQUESNE LIGHT COMPANY, COLFAX STATION. FRONT WALL, BOILER NO. 16, AFTER CHANGE IN AIR DISTRIBUTION

Service hours since last cleaned, 1200.

FIG. 20 DUQUESNE LIGHT COMPANY, COLFAX STATION. BOILER NO. 16, TUBE BENT BY FALLING SLAG

Service hours since last cleaned, 1200.

The illustrations, Fig. 8 to 20, show clearly the furnace conditions in two quite different types of installation. One case shown is installed in the steam-heating plant of the Allegheny County Steam Heating Company, and the other in the Colfax Station of the Duquesne Light Company. The chief characteristics of these two installations are as follows:

	Colfax Station	Allegheny
Boiler surface, sq. ft.	27,680	29,884
Screen surface, sq. ft.	927	1,600
Total surface, sq. ft.	28,607	31,484
Furnace volume, cu. ft.	17,600	20,500
Number of burners	14	16
Maximum output guaranteed, lb. steam per hour from and at 212 deg. fahr.	320,710	412,400
Secondary air temperature, deg. fahr.	250[1]	70[1]

[1] Nominal temperatures.

The Colfax boilers have been operated with an actual output of 304,000 lb. of steam per hour, and the Allegheny boilers with actual outputs of 340,000 lb. of steam per hour. Only minor difficulties have been experienced with the Allegheny boilers such as are met with in starting up any new equipment. The case at Colfax is quite different. While these boilers have been in regular service since their installation, the furnace conditions have been far from satisfactory. Difficulty from slag adhering to the walls has been serious and the writer believes that the impracticability of using, with pulverized fuel, the ordinary type of air-cooled setting with preheated air has been fully demonstrated. The fuel used at Colfax is not the best as regards its slagging properties, but is about an average.

A. R. Mumford.[1] The thoughts expressed in this paper are so clearly stated that they enable the ordinary engineer to obtain a more complete conception of the progress of combustion in a pulverized-coal furnace.

The conclusion drawn from this paper by the writer is that radiation from the flame in a pulverized-coal furnace has no great effect on ignition or combustion because of the fact that such heat as is radiated is obtained principally from the envelope of the flame. Radiation from the interior parts of the flame is prevented by the interposition of coal or carbon particles between the source of radiation and the suggested water-cooled walls. The conclusion can be drawn, however, that any reduction in the temperature of the envelope of a powdered-coal flame causes heat to be radiated to the envelope from any hotter particles in the interior so that the heat radiated to the water-cooled walls is eventually, although not originally, drawn from the whole flame. If the above conclusions are drawn correctly, it appears that the whole furnace condi-

[1] Fuel Engineer, New York Steam Corporation, New York, N. Y. Assoc.-Mem. A.S.M.E.

tions are affected by water-cooled walls and radiation can take place from the interior of the flame to the walls by an indirect path.

It is interesting also to follow the effect of radiation on ignition. As the author points out, the outer envelope of a pulverized-coal stream as it enters the furnace receives heat by radiation from the burning particles in other parts of the furnace, as well as from any incandescent brick which may be present. When this envelope is heated to the ignition point and combustion begins, then it re-radiates heat to the water-cooled walls and also radiates heat to the cooler coal particles in the center of the stream. These coal particles in the interior of a stream of pulverized coal can receive heat only by radiation from the outer envelope next to them or from the ignited coal further along in their path, because of the interposition of a screen of solid particles between it and other sources. Therefore it seems that any loss of heat from a powdered-coal flame by radiation to cool surfaces must not only reduce the temperature of the envelope but also the temperature of the whole flame, even if only to a slight degree. It seems probable, also, that any reduction of flame temperature will reduce the radiation of heat to the particles awaiting ignition, so that ignition ought to be measurably slower in a water-cooled furnace than in a refractory furnace.

The writer is an advocate of water-cooled walls because of their obvious advantages structurally and from the standpoint of the reduction of the radiation and convection losses from the external surfaces of the setting. He feels that water-cooled furnaces will affect a pulverized-coal flame adversely and that this adverse effect can be overcome by a study of another set of conditions not mentioned by the author. These conditions have to do with the proper relation of the quantity of primary air to coal in order to maintain the maximum rate of flame propagation under any given set of conditions.

It has been shown elsewhere that a high-volatile bituminous coal requires less careful regulation of the quantity of primary air to maintain the maximum rate of flame propagation than does a low-volatile anthracite. This fact indicates that coal in the pulverized form can be considered much as the gas-air mixture of W. R. Chapman in his work on The Propagation of Flame in Mixtures of Ethylene and Air, and that unsatisfactory combustion conditions will obtain where the primary air is not under separate and careful control.

John Blizard's valuable mathematical study of the Terminal Velocity of Particles Flowing in a Viscous Fluid is also of importance in any consideration of combustion in pulverized-coal furnaces. The fact that the light, small particles are sifted from the coal stream and are forced to take the short path out of the furnace indicates that the radiation to the remaining heavy, large particles

must be greater in order to maintain rapid ignition, or that the quantity of primary air must be so controlled that the maximum speed of flame propagation will be maintained.

J. GOULD COUTANT.[1] Having read these papers[2] with a view to practical application, the writer has arrived at a conclusion similar to that of Dr. Lucke, who said: "No problem of equal practical importance is more difficult or elusive than the prediction of fuel effect by calculation, or the determination of suitable fuel for a given purpose."[3]

The possibility of using the data for making calculations in regard to the process of combustion, furnace or boiler, depends on skill in collecting such necessary data as can be obtained by observation or measurement. If ever the rule of thumb is to be replaced by scientific methods, the change must be based on experiment, classification of results, and the calculation therefrom. Every problem thus is an exercise in pure logic, supported by mathematical reasoning. The premises are observed and accepted facts, which are to be found in the methods and principles now used for conducting boiler trials. These form a basis sufficiently accurate for the design of boiler furnaces.

The use of pulverized fuel with 60 per cent of the total ash content arriving at the first gas passage of the boiler is dependent on the temperature at which the ash will melt, i.e., coal should be burned between the temperatures of 1950 deg. fahr. and the point at which the ash that is held in suspension in gas currents becomes liquid. Thus boilers burning coals that contain ash melting at 2500 to 2600 deg. fahr. will have twice the range for steaming capacity as when burning a fuel having an ash that will become liquid at 2100 deg. fahr. Thus a boiler burning coal containing ash that will melt at 2600 deg. fahr. may be operated at 400 per cent of rating, while the same boiler burning coal with ash that melts at 2100 deg. fahr. will have a maximum capacity of 200 per cent of rating, at which rates of boiler evaporation the hard ash will begin to form in tubes and obstruct the draft. This is true unless a constant temperature furnace is used.

I — The first determination is that of the temperature at which ash taken from the tubes of the boiler will melt and run, as this is the maximum of ruling temperature at which the gases can arrive at the boiler tubes. In all powdered-coal-fired furnaces it is intended that all the heat in the fuel should be liberated within the combustion chamber before the products of combustion arrive

[1] Furnace Engineering Company, Inc., New York, N. Y. Mem. A.S.M.E.

[2] This discussion also refers to the paper, Radiation in the Pulverized-Fuel Furnace, by W. J. Wohlenberg and D. G. Morrow. See p. 127.

[3] Engineering Thermodynamics, 1913, pp. 646.

at the first pass of the boiler. The basis for this thermic study is the knowledge of heats of combustion.

The amount of heat in fuel as determined in the bomb calorimeter, when liberated in the combustion chamber, is taken up by the products of combustion, with only 1 or 2 per cent being lost by radiation from furnace walls. The specific heats and thermal capacities of gases that we now accept were determined by Mallard and LeChatelier, and will serve as a means to calculate the theoretical temperatures of gases arriving at the tubes in the first pass of the boiler.

II — The second determination is that of the theoretical temperature of the gases arriving at the boiler tubes.

Two conditions have been satisfied, based on accepted data. The next of relative importance is the reduction of the theoretical gas

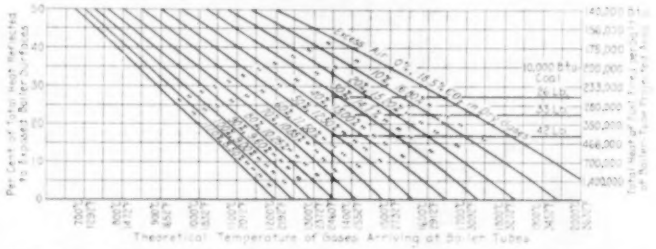

FIG. 21 THEORETICAL TEMPERATURE OF GASES IN BOILER FURNACES

Pittsburgh Coal—13870 B.t.u. (7704 cal.)

Analysis (Per cent)	Proximate Analysis (Per cent)	Percentage Composition (By volume)
Carbon, 77.00	Moisture, 0.55	CO_2, 17.00
Hydrogen, 5.35	Volatile, 37.00	H_2O, 7.06
Nitrogen, 1.58	Fixed carbon, 56.51	SO_2, 0.10
Sulphur, 1.17	Ash, 5.94	N_2, 75.84
Oxygen, 8.96		
Ash, 5.94		

In practice, deduct for 45 per cent humidity 100 deg. cent. (180 deg. fahr.); also furnace radiation, 50 deg. cent. (90 deg. fahr.); total deductions, high-temperature furnaces, 150 deg. cent. (270 deg. fahr.); boilers and other furnaces, 100 deg. cent. (180 deg. fahr.).

temperature to that of the melting point of the ash or actual gas temperatures.

The gases, moving rapidly in space, transfer heat by convection and conduction to the particles held in suspension. It is possible that a small amount might leave by radiation. The particles or globules then become like little suns radiating heat from one to another, to the side walls, arches, and refractory surfaces, until the reflected heat comes in line with the boiler surfaces.

It is improbable that any means will be devised for calculating the radiation of heat, as such calculation would be furthermore upset by the fluctuations of fuels, ash accumulations, operating conditions, composition and density of gases, etc.

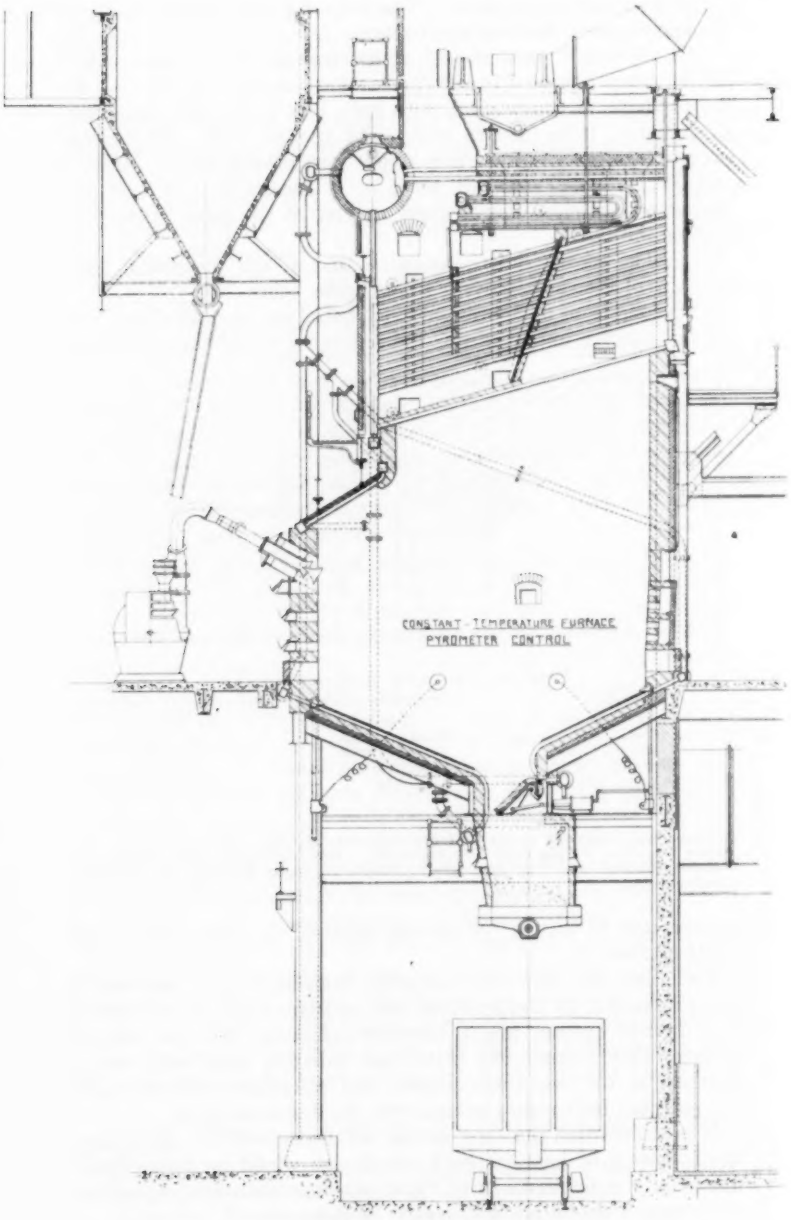

FIG. 22 CONSTANT-TEMPERATURE FURNACE

In view of the elusive character of calculations, based on radiation, it is much more logical to adhere to the actual results and data that can be obtained from the thousands of boilers in operation. The writer has visited more than twenty power stations and compared them on the basis of the following data, and has determined that the maximum absorption of heat per square foot of projected area of boiler surface per hour is 70,000 B.t.u.

TABLE 1 CONSTANT-TEMPERATURE FURNACE

COAL ANALYSIS, PER CENT:

Fixed carbon, 52.5.
Hydrogen, 3.5.
Oxygen, 9.1.
Nitrogen, 1.4.
Sulphur, 3.5.
Ash, 15.0
Water, 13 to 15.
Not less than 10,000 B.t.u. as fired.
Ash softening point, 1950 deg. fahr., assumed.
Ash running point, 2180 deg. fahr.
Gases at boiler tube, 2180 deg. fahr., maximum.
Boiler-tube surface exposed to fire, 288 sq. ft. projected area.
Furnace volume, 9600 cu. ft.

1500-Hp. B. & W. Cross-Drum Boiler

16 kw-hr. per ton (2000 lb.) operating at full capacity of pulverizer
19 kw-hr. " " " " " " " " "
24 kw-hr. " " " " " " " " "

Simplex units, 12,000 lb. capacity each, No.	Power for preparation and burning, per cent, 18,000 B.t.u. per kw.	Evaporation, from and at 212 deg. fahr., lb.	Rate of driving, per cent of rated capacity	Boiler efficiency, per cent	Coal burned per hour, approx. lb.	CO_2 approx., per cent	Excess air, approx., per cent	Temp. of gases at tubes, deg. fahr.	Coal burned per sq. ft. projected area (10 per cent moist.), lb.	Temp. of gases leaving boiler, deg. fahr.
1	2.15	64,680	125	82	7,680	14	30	2180	26	460
1	1.7	77,625	150	80	9,150	13	40	2180	32	465
1	1.45	90,555	175	74 approx.	11,700	12	50	2180	40	525
Projected Area Boiler Tubes 288 + Water-Tube Floor 192 = 480 Sq. Ft. Projected Area										
1	1.7	77,625	150	82	9,100	14	30	2180	Floor partly covered with ash	465
1	1.54	90,555	175	81	10,900	14	30	2180	"	475
1	1.45	103,500	200	79	12,650	14	30	2180	26	485
2	2.0	129,370	250	78	15,840	13	40	2180	33	520
2	1.63	155,250	300	74	20,400	12	50	2180	42	570

[1] NOTE.—More tube surface could be placed in floor.

III — The third determination is that of the actual temperature of gases arriving at boiler tubes, which can be found graphically by referring to the curve, Fig. 21. In order to thoroughly understand this, assume, for example, the constant-temperature furnace, Fig. 22, burning coal at the fixed rate of 10,000 to 35,000 B.t.u. per cu. ft. of furnace, the special features of which enable a boiler to be operated at the maximum steam capacity, say, of 300 per cent of rating, regardless of the kind of coal, melting point of ash, or preheat of the air.

The constant temperature of the gases arriving at the boiler tubes is maintained by the control of the water-tube surfaces exposed to the flame, in proportion to the amount of coal burned per square foot of projected area of tube surfaces exposed; i.e., the ash is allowed to accumulate on the cast-iron floor during light loads, or is swept clean at heavy loads, being controlled by a pyrometer operating a control steam valve.

Referring to Table 1, it will be seen that the ash in coal melts at 2180 deg. fahr. This is the actual temperature of the gases at the tubes; the theoretical temperature can be arrived at as follows:

	Deg. fahr.
Temperature at which ash melts, actual temperature of gas.....	2180
2 per cent radiation losses reduce theoretical temperature approximately	80
45 per cent humidity reduces theoretical temperature approximately	120
10 per cent moisture in coal, which is in excess of that for which curve is calculated, reduces theoretical temperature approximately	80
Theoretical temperature of gases.....	2460

Referring to Fig. 21, and noting that the temperature 2460 deg. fahr. line cuts the CO_2 curves, the following calculated results are shown in Table 1:

Curve 14 per cent CO_2 reads to right, 26 lb. of coal per hour per sq. ft. projected area.
Curve 13 per cent CO_2 reads to right, 33 lb. of coal per hour per sq. ft. projected area.
Curve 12 per cent CO_2 reads to right, 42 lb. of coal per hour per sq. ft. projected area.

The boiler, Fig. 22, can be operated at 75 to 200 per cent of rating, maintaining a constant temperature or by increasing tube surface in floor to 300 per cent of rating. For higher rates of evaporation, it is necessary to control the amount of excess air, which may be done automatically by a pyrometer working in conjunction with the motor on control steam valve.

The curves have been calculated on the basis of the thermal capacity of gases and 70,000 B.t.u. exchanged by radiation per square foot of projected area per hour.

The development of temperature control for automatic temperature regulation between the temperatures of 1800 deg. fahr. to 3200 deg. fahr. — the pyrometer regulator shown in Fig. 22 maintains the temperature of the furnace at the maximum burning efficiency of fuel — makes exact calculation of the furnace unnecessary when the constant-temperature furnace is used. Further, the burning efficiency is controlled automatically and is maintained.

THE AUTHOR. Professor Wohlenberg has discussed whether it is re-radiated heat or reflected heat from the brick. It is probable that both of us are attempting to say the same thing. Both he and Mr. Bailey discussed the effect of the furnace walls on the rate of radiation of the coal particles. There is one other point that has to be taken into consideration besides the effect of the coal particle itself, and that is the CO_2 and water vapor which result from the combustion of the coal. Both have considerable radiant-heat-absorbing capacity and radiant-heat-dissipating capacity, whereas oxygen, nitrogen, and the other gases do not. The presence of these around the coal particles must have a considerable effect on the rate of radiation, even though it may be in the non-luminous section of the furnace. The author believes that the effect of that is not yet fully understood, although Professor Wohlenberg discusses it in his paper.

The Fuller pot furnace described by Mr. Brooks is a promising development of great interest, and the attempt there to secure the turbulence which the paper suggests is a real step in the right direction.

Mr. Clarke raises the question of the effect of surfaces in furnaces on heat transferred. The author agrees with this portion of his discussion. Mr. Clarke disagrees on the effect of the color of the water-cooled surfaces and the control of that color in the furnace. It seems that in the newer furnaces there is little adherence of slag and ash to the furnace walls, and, therefore, if these are colored in some way by the aluminum coating that has been suggested, they will not be covered up by the slag and hence should have some effect on furnace temperatures. While we have no positive knowledge as yet, nevertheless in the heating surfaces where these calorized materials have been used they do not change color to any appreciable extent after the first oxidation. This was what was in mind in suggesting that in the pulverized-coal furnace. The idea would be to attempt to apply the English method of applying aluminum coating to the tubes rather than to calorize the whole tube. All of the discussions indicate that intensive thought is being given to this problem.

No. 1956

RADIATION IN THE PULVERIZED-FUEL FURNACE

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and

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Non-Member

This paper deals with the fundamentals concerning radiation in the pulverized-fuel furnace. It is shown that the radiating power through the flame surface depends primarily on the size of the particle and that based on the flame-surface area, its magnitude relative to black radiation is low. The heat-absorption intensity at the cold surface, however, may be considerably higher than the relative radiance of the flame. This factor depends on both the amount and the disposition of the refractory furnace lining. Application of the method outlined enables, by an analytical process, quantitative evaluation of the furnace energy process, and the data yielded are of such a nature as to indicate by comparison which furnace proportions will give the best results.

USUALLY the subject of radiation within the boiler furnace is dismissed with a statement of the difficulties involved in its accurate evaluation by any analytical method. Sometimes certain approximations are made by assuming, for easily workable cases, fuel surfaces parallel to the exposed surface. Generally black-body conditions are used in the computations. Such factors as reflection and radiation from the refractory, interference offered by the gaseous and solid particles of the flame, and certain other influences are neglected, with the result that, in general, a rather hazy notion is had of the detailed energy process involved.

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[2] Yale University.

Acknowledgment. The authors wish to express their indebtedness to Mr. A. B. Stickney II, a graduate student at Yale, for suggesting a number of valuable devices which have been of great service in working out this problem.

Contributed by the Fuels Division and presented at the Spring Meeting, Milwaukee, Wis., May 18 to 21, 1925, of The American Society of Mechanical Engineers.

2 In this paper an attempt is made to solve by analytical methods the energy process occurring in the boiler furnace. The general theory is applicable to any boiler furnace, but special devices have been included for its particular application to the pulverized-fuel furnace. The limitations as to exact definitions of the various conditions involved result in certain imperfections as to details, but that a thorough study may be of more than pure academic value is shown by the kind of information which is yielded.

3 In applying such a method the uninitiated should, however, be warned that it is no more than a tool which will fashion a product whose value is in accordance with the skill and understanding with which it has been used. This aspect is all the more important because of the foregoing assumptions, admittedly somewhat imperfect, which the authors have made in evaluating certain conditions. Significant in this respect are the assumed dimensions of the temperature quantities evaluated, no attempt having been made to account for the momentary and zonal variations. The temperature quantities are thus mean radiational values. Their relation to actual conditions is later discussed.

4 The paper is divided into four parts. The first part contains the results of an application of the method to a few cases. It is not intended, however, that this part of the subject shall be treated exhaustively at this time. Attention is now called rather to the method and general type of results, leaving a more comprehensive application for a later date. The second, third, and fourth parts include, respectively, the theoretical development, sample calculations, and appendices, the latter containing material used and certain theoretical developments.

PART I

Type of Results Obtainable

5 Some of the significant results yielded by the application of such a theory may be enumerated as follows:

a Radiating mean value of the flame temperature, T_U.

b Radiating mean value of the refractory temperature, T_R.

c Radiation intensity of the flame surface compared to that of a black body. This is called the relative radiance, μ_1.

d The radiation efficiency of the furnace compared to that furnace in which the cold surface is immediately in front of a perfectly black fuel bed at the temperature T_U. This ratio is called the radiation efficiency of the furnace and is designated by the symbol μ_2. It is a measure of the radiant and reflected energy which actually strikes the cold surface as compared to that which would strike it with a perfect black fuel bed immediately in front of it.

It does not include the heat absorption by convection at cold surface and does not correct for the reflection from the cold surface.

e The total heat absorption at the cold surface compared to that which would result by means of radiation only in a furnace with a black fuel bed immediately in front of the black tubes. This quantity is called the relative absorption, and is represented by the symbol μ_3.

f The fraction of liberated energy which is absorbed in the furnace. This quantity is called the furnace absorption efficiency and is designated by the symbol μ_4.

g The influence of furnace volume, shape, and disposition of cold surface on the foregoing quantities.

h The influence of rates of energy liberation on the foregoing quantities for a given furnace.

i The influence of fineness of pulverization on certain of the foregoing quantities.

6 Because of the limited time available since the work was started it has been possible to evaluate quantitatively the conditions in a cubical furnace only. The cold- and refractory-surface arrangements considered are designated as follows:

Arrangement A. Cold surface at the top face only which forms the aperture for the escaping gases.

Arrangement B. Cold surface at the top and bottom faces, the aperture for the escaping gases being at the top.

Arrangement C. One-half of the interior surface as cold surface, distributed so that one-half of the top, bottom, and two sides, and total furnace back is cold surface.

Arrangement D. Top, bottom, and two opposite sides as cold surface.

Arrangement E. Furnace totally enclosed by cold surface.

7 Table 1 includes the figures arrived at for the various

TABLE 1 FOR PARTICLES OF 0.002 IN. DIAMETER AND T_C = 440 DEG. FAHR.
[Illinois Coal (see Part III, p. 152) pulverized 75 per cent through 200-mesh, burned with 30 per cent excess air]

Type of Furnace	Energy Liberation, B.t.u. per cu. ft. per hr.	Arrangement of Exposed Surface	Approx. Mean Flame Temp. T_U Deg. Fahr.	Approx. Mean Refractory Temp. T_R Deg. Fahr.	μ_1	μ_2	μ_3	μ_4
Cubical Furnace 20′×20′×20′ with Flame 19′×19′×19′	20,000	*A*	2555	2275	0.36	0.84	0.77	0.30
		B	2320	1890	0.38	0.68	0.66	0.37
		C	2270	1925	0.38	0.49	0.50	0.39
		D	2150	1580	0.40	0.48	0.48	0.43
		E	2060	....	0.41	0.38	0.40	0.45
	15,000	*A*	2420	2165	0.37	0.85	0.78	0.34
		B	2185	1790	0.39	0.69	0.67	0.41
		C	2120	1810	0.40	0.51	0.51	0.43
		D	2010	1475	0.41	0.49	0.49	0.46
		E	1920	....	0.41	0.37	0.41	0.49
	10,000	*A*	2225	1990	0.39	0.87	0.81	0.40
		B	1980	1630	0.41	0.72	0.70	0.46
		C	1915	1645	0.42	0.53	0.54	0.48
		D	1810	1340	0.43	0.50	0.52	0.51
		E	1730	....	0.44	0.40	0.43	0.54

surface arrangements at different rates of energy liberation. Some of these data and certain other factors are represented graphically in Figs. 1, 2, 3, 4, 6, and 7.

Discussion of Results

8 *Comparison of Furnace Types.* As an example, furnace B will first be compared with furnace C. It is to be noted that C contains considerably more cold surface than B but that the flame temperature of C is only slightly lower than that of B, and, contrary to expectations, the refractory temperature of C is actually higher than that of B. The result is only a slightly greater heat absorption (compare values of μ_4) by C for a considerably increased cold surface. This condition exists because in C the cold surface "sees" other cold surface through a much greater solid angle than it does in B. Likewise the refractory "sees" other

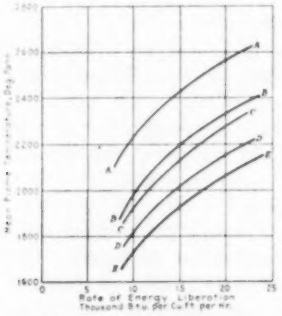

Fig. 1 Variation of Mean Flame Temperature with Rate of Energy Liberation

refractory through a larger solid angle. The net heat absorbed from the refractory radiation and reflections by the cold surface is thus considerably less than in B in spite of the increased cold surface. The disposition of the cold surface and furnace shape thus has a very marked influence on its effectiveness for absorbing heat radiations from the refractory.

9 The decrease in the effectiveness of the surface for the variation in the arrangements considered may be shown by evaluating the ratio

$$\frac{\mu_4}{\text{exposed-cold-surface fraction}}$$

The result is shown in Fig. 4. This curve should not, however, be considered as generally applicable with accuracy. It fits exactly only the conditions for which it is computed. If, for instance, a furnace of the type $A = B = nC$ is considered, in which n is, say, 3 or 4, it becomes obvious for arrangement C (50 per cent cold sur-

face disposed as shown in Fig. 5) that both the cold and refractory surfaces "see" considerably less of like surfaces and hence considerably more of the other. The cold surface will therefore be more effective.

10 *Influence of Energy-Liberation Rate.* Inspection of Fig. 3 shows that, although the coefficients μ_1, μ_2, and μ_3 do not change greatly, the fraction of liberated energy μ_4, which is absorbed in the furnace, decreases rapidly with increase of energy liberation. This, of course, means that a larger part of the liberated energy must be absorbed in the convection zone of the boiler and hence that the efficiency must necessarily decrease, which is otherwise obvious by inspection of the flame-temperature curves, Figs. 1 and 2. The general relation between efficiency and energy liberation is, of course, well known, but the exact nature of the detail process involved is perhaps not so well known.

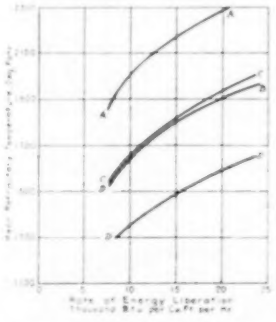

FIG. 2 VARIATION OF MEAN REFRACTORY TEMPERATURE WITH RATE OF ENERGY LIBERATION

11 It should be noted in this respect that the furnace volume has been considered fairly well filled with flame for all rates of energy liberation, whence it follows that the furnace designed for a low energy-liberation rate should, under such conditions, yield with the same total heat-absorbing surface a higher efficiency, other conditions being equal. The influence of a reduced flame volume as a result of decreasing the energy-liberation rate has not been shown.

12 *Other Black-Body Comparisons.* Of particular importance is the disclosure that the relative radiance μ_1 of the flame cannot, for average conditions, be equal to that of a black body. This condition is, of course, reflected also in reduced values of the radiating and absorption efficiencies. Calculations based on black-body assumptions thus result in an error whose magnitude is high or low, depending upon the fraction of the envelope which is composed of refractory surface. For a large fraction of refractory

surface the re-radiated and reflected energy adds to the direct radiations discharged from the flame to the cold surface, but even here it appears that an error of 20 to 30 per cent is easily possible.

13 For a furnace completely filled with flame the values of μ_1 and μ_2 should agree when the cold envelope is complete. The

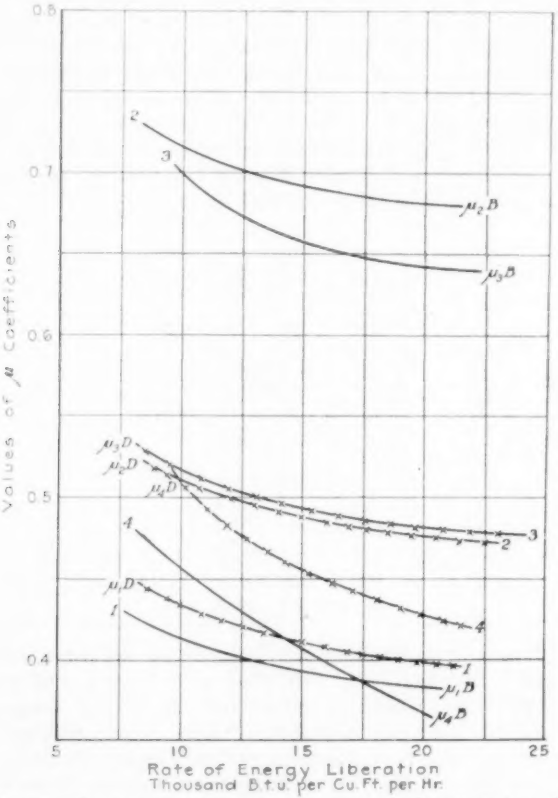

FIG. 3 COMPARISON OF μ_1, μ_2, μ_3, AND μ_4, CASES *B* AND *D*

difference appearing for cases *E* in this respect is due to the clearance assumed between the flame surface proper and the furnace walls, the flame volume being assumed so as to take up approximately 85 per cent of the total furnace volume. The differences respectively between μ_3 and μ_4 and μ_1 are still larger for these conditions, due to the heat absorbed by convection at that exposed cold surface not at the aperture, and it is interesting to note that for the lower rates of energy liberation as much as

one-fifth of the heat absorbed in the furnace may be absorbed by convection at the cold surfaces. Thus for E with 10,000 B.t.u. energy liberation, $\mu_1 = 0.44$ and $\mu_4 = 0.54$. The difference must obviously be due to the convectional heat. This is perhaps obvious when noting that the radiation mean of the flame temperature T_U is only 1730 deg. fahr.

14 By reference to Fig. 3 it is seen that for furnace B, $\mu_2 > \mu_3$, whereas for D the reverse is true. This is due to the increased absorption by convection at the cold surface in D as compared to B. Reference to the definitions of these coefficients will make this clear.

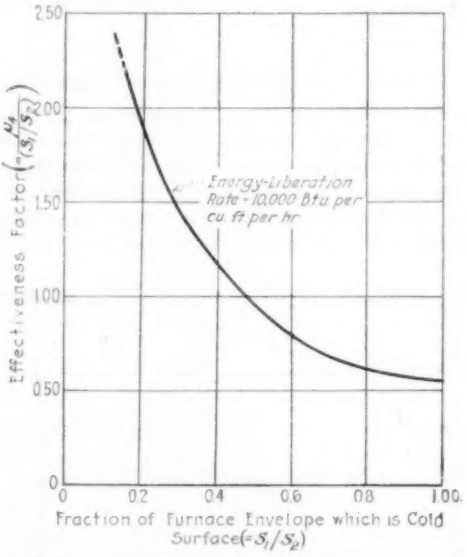

FIG. 4 VARIATION OF EFFECTIVENESS FACTOR AND FRACTION OF FURNACE ENVELOPE WHICH IS COLD SURFACE

15 *Influence of Pulverization Fineness and Flame Volume on Relative Radiance.* The curves of Figs. 6 and 7 illustrate the variation of the relative radiance with flame and particle size. For the average particle size as determined when 75 per cent of the sample is passed through a 200-mesh screen, the variation of relative radiance is not large until the flame becomes comparatively small. This is seen by reference to Fig. 6. Fig. 7 indicates the variation of flame radiance with fineness of pulverization. It should be noted that the radiance improves appreciably as the particle is reduced in size.

16 For very large particles the radiance of course is materially reduced because of the now decreased surface area on the relatively few particles in suspension, and eventually the influence of particles becomes small even as compared to the radiating portions of the gaseous constituents.

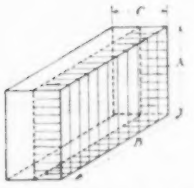

FIG. 5 ARRANGEMENT OF COLD SURFACE REFERRED TO IN PAR. 9

17 Other interesting relations will become apparent from further study of the results and method, but the discussion is assuming too great a proportion. A review of the assumptions on which the results are based is, however, of sufficient importance to warrant its inclusion at this time.

CRITICAL REVIEW OF ASSUMED CONDITIONS

18 Since a uniform distribution of the fuel particles has been assumed, it follows that for the total flame the energy transfer by radiation is a maximum. In present furnaces the particles are blown to the flame as a dense central cloud along the flame axis, which later on becomes more or less rapidly distributed throughout the flame mass. As compared to that in a uniform mixture

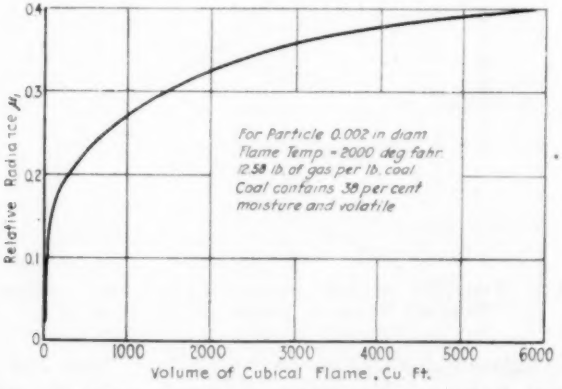

FIG. 6 VARIATION OF RELATIVE RADIANCE WITH FLAME SIZE

the actual radiation interference fraction, or shielding coefficient, will thus be large in the flame zone near the burner. The total radiance from the flame is therefore decreased, but the relative radiance through the flame surface immediately surrounding the entrance zone is increased because of the increased particle congestion. This combined with the decreased rate of energy liberation, due partly to the necessity of raising the incoming mixture

to the ignition point and also to the high over-richness of the mixture, results in a comparatively low flame temperature in what may be called the entrance or ignition zone of the flame.

19 As the gases travel further along the flame path the fuel-air distribution is better, and at the point of change of direction the turbulence or relative motion between fuel particles and surrounding air becomes greatly increased with a resulting maximum rate of energy liberation, so that, in spite of the decreased shielding of

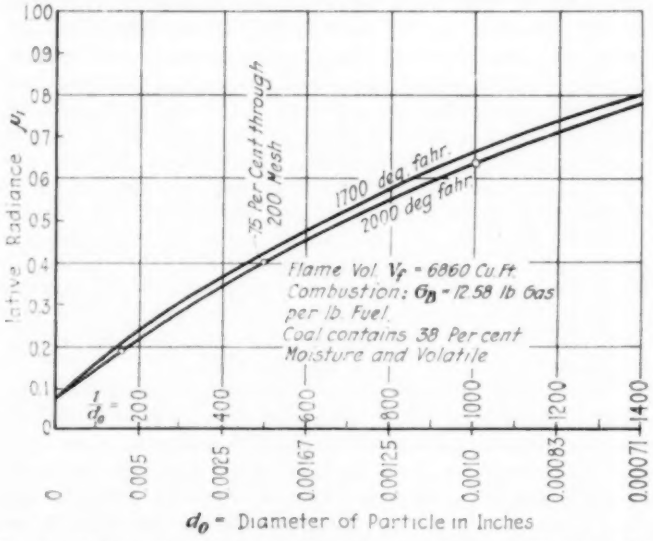

FIG. 7 VARIATION OF RELATIVE RADIANCE WITH FLAME-PARTICLE SIZE

the particles, the flame temperature is a maximum.[1] The radiation discharge from this zone is probably likewise a maximum.

20 The surface oxygen intensity (pounds of oxygen combining with carbon per hour per unit surface exposed by particles) has been assumed constant as the particle travels through the furnace. Actually this factor is undoubtedly greatest where the gas travel changes in direction. The value used is thus a mean value based on the mean rate of energy liberation and its inclusion in this manner should yield results representative of mean conditions.

21 In the present investigation the mean solid angles for radiation have been measured from the center of volume of the furnace. Perhaps a more accurate solution would result by determining first the location of the center of energy liberation and the center

[1] See an investigation of Powdered Coal as Fuel for Power Plant Boilers, by Kreisinger, Blizard, Augustine, and Cross. Bur. of Mines, Bull. No. 223.

of particle distribution for the purpose of finding a better mean point from which to measure the radiation angles and also for determining the shielding coefficients. The latter have been assumed to have the same mean value for points radiating from the refractory to the cold surface, from the flame to the refractory, and from the flame to the cold surface. They have been computed as shown in Appendix No. 4 for the mean "seeing" condition of particles within the flame, and may thus actually have values slightly different from those assumed when considering radiations from the refractory to the cold surface. Such differences are, however, probably within the error of other necessary assumptions.

22 The shielding coefficients as used assume zero radiation from the shielded to the shielding parts of the overlapping surfaces. Actually the shielding particles will have their temperature reduced by radiations to the furnace walls. The shielded particles will therefore have a somewhat higher temperature, as a result of which they will radiate to the shielding surface. Because of their reduced temperature the latter will thus radiate less energy than they are charged with by present assumptions. On the other hand, the shielded particles will radiate more energy because the shielded parts have been assumed as transferring zero energy. The sum total of the actual energy transfer therefore probably closely approximates that found on the assumed basis.

23 Since the flame temperature T_U represents a radiation mean, the temperatures in an actual furnace should vary from something above this value at the center of energy liberation to some value below it at the outer flame surface. This, however, as before mentioned, does not necessarily result in unreliable values for the coefficients μ_1, μ_2, μ_3, and μ_4. These are likewise radiation means and as such should, by application, yield reasonably correct quantitative values for the mean rates of energy transfer. Similar remarks apply to the radiation mean temperature value T_R.

Utility of Results

24 Perhaps the most useful data, if correct, yielded by such an investigation are contained in the values of the coefficients μ_1, μ_2, μ_3, and μ_4. It is obvious that their magnitudes are indicative of the effectiveness of both the flame and furnace as a partial means of transferring liberated energy to the water and steam. The relative radiation values of various furnace types and surface arrangements apparently can be evaluated with reasonable accuracy in such terms, even if the various temperature magnitudes arrived at do not describe in sufficient detail the actual flame as to temperature dimensions. These coefficients are therefore of use to the designer of heat-absorbing vessels in evaluating the convection zone, as well as in evaluating the furnace conditions.

PART II

General Method

25 The energy equation of the boiler furnace may be stated in the form:

$$dQ_U/dt = dQ_B/dt + dQ_F/dt \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

The first term, dQ_U/dt, represents the time rate of energy liberation by burning fuel in the furnace. Its magnitude in B.t.u. per hr. is expressed by the product

$$G_U U \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [2]$$

in which U represents the B.t.u. actually liberated per pound of fuel fired and G_U the pounds of fuel fired per hour. If the fuel is completely burned, U represents its calorific value. A high temperature may limit the reaction, but in this respect conditions in the boiler furnace are such that the driving force of the reaction is sufficient to complete combustion unless the gases are prematurely cooled by contact with cold surfaces.

26 The second term, dQ_B/dt, represents the rate at which *liberated energy* is absorbed by the *burned gases* in raising the temperature from initial conditions to the flame temperature. Its magnitude is the result of heat addition at constant pressure to each of the constituents forming as a whole the burned-gas mixture. Symbolically this quantity may be expressed in the form

$$G_{CO_2}\Delta h_{CO_2} + G_N \Delta h_N + G_O \Delta h_O + G_{SO_2}\Delta h_{SO_2} + \quad . \quad . \quad . \quad [3]$$

in which G_{CO_2}, G_N, G_O, etc., represent the weights, lb. per hr., of the constituents CO_2, N_2, O_2, etc., and the multiplying terms Δh_{CO_2}, Δh_N, etc., the heat capacities at constant pressure between the initial temperature T_1 and the flame temperature T_U. Usually these quantities are expressed merely as the integral

$$\left[\Delta h\right]_1^2 = \int_1^2 C_m dt$$

in which C_m represents the mean specific heat for the temperature range in question. A more accurate method, however, employs the heat-capacity curve plotted against temperature as outlined in Appendix No. 1. The sum of the products $G\Delta h$ then yields quite accurately the energy absorption rate dQ_B/dt by the burned gases.

27 The last term dQ_F/dt represents the time rate of energy loss from the solid and gaseous portions of the flame during their travel through the furnace. This term has the general dimensions

$$dQ_F/dt = K(T_U^n - T_C^n) + Z \quad . \quad . \quad . \quad . \quad . \quad . \quad [4]$$

in which K is a constant, T_U the absolute mean radiating temperature of the hot body, and T_C the absolute mean radiating temperature of the cold body. With $n = 4$, the term $K(T_U^4 - T_C^4)$ represents the Stefan-Boltzmann law, in which form it is applicable

within the limits of error involved in the definition of other conditions. The term Z denotes the heat transferred by convection as the gases flow along the furnace walls. It may be otherwise represented as

$$Z = \Sigma S z \Delta t \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [5]$$

in which S represents furnace-wall surface, z the coefficient of heat transfer in B.t.u. per hr., and Δt the mean temperature difference between the furnace wall and the layer of moving gas immediately in front of the surface. The constant K depends on a great many factors and its evaluation presents one of the chief problems of this paper.

28 Both the solid fuel portions and the gaseous flame masses radiate to both the refractory and exposed boiler surfaces. The transfer of heat by convection occurs also at all of the walls to a greater or less extent, and the process is further complicated by reflection from the various surfaces. Fig. 8 is a graphical representation of such an activity resolved into those net effects in terms of which the general equations may be stated.

29 The total energy as heat supplied to the furnace is thus divided into four parts, such that

$$Q_T = Q_U + Q_{AR} + q_{CR} + q_{CT} \quad . \quad . \quad . \quad . \quad . \quad . \quad [6]$$

in which Q_U = energy liberated by burning the fuel

Q_{AR} = energy transferred from the furnace walls to the air supplied for combustion

q_{CR} = that part of the radiated energy $(Q_{RC} + q_{RC})$ discharged from the refractory R which strikes the cold surface C and is reflected back into the furnace

and q_{CT} = that part of the energy Q_{TC} discharged from the flame mass which by direct radiation strikes the cold surface and is there reflected back into the furnace.

It is to be noted that the reflected energy $(q_{CR} + q_{CT})$ and the energy Q_{AR} are recirculated within the flame mass, thereby increasing the net energy Q_{TB} absorbed by the burned gases, which, as a result, are discharged at a higher temperature, through the aperture leading from the furnace into the boiler convection zone. Of the heat $Q_{TR} + Z_R$ discharged from the flame and fuel mass to the refractory R, a certain portion X_R is lost by conduction through the walls to the surrounding atmosphere. Another portion Q_{AR} is sometimes, as shown, conducted to the incoming air, thereby raising its temperature. A third part q_{RC} represents the *net reflection* to the cold surface, and the remainder Q_{RC} is absorbed by the wall and re-radiated to the cold surface.

30 The net heat loss Q_F from the flame mass is thus a sum of the quantities

$$X_R + Q_{CR} + Q_{CT} + Z_C \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [7]$$

in which X_R has been defined, and Q_{CR} and Q_{CT} denote respectively the heat absorptions at the cold surface by virtue of radiations and reflections from the refractory to the cold surface, and radiations from the flame and fuel mass to the cold surface. The last term Z_C indicates the energy absorbed by convection at the sum total of the exposed cold surface in the furnace which is *not* located at the aperture through which the gases escape into the principal convection zone of the boiler. Heat absorbed by con-

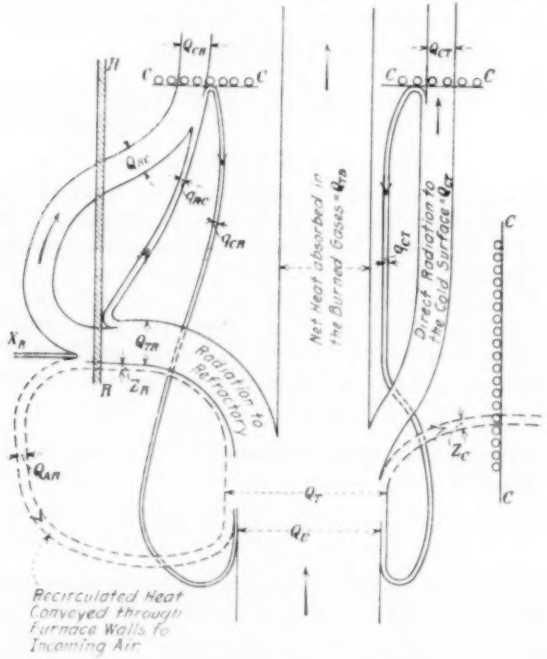
Net Heat absorbed in the Burned Gases = Q_{TB}
Direct Radiation to the Cold Surface = Q_{CT}
Radiation to Refractory
Recirculated Heat Conveyed through Furnace Walls to Incoming Air

FIG. 8 FURNACE ENERGY DIAGRAM

vection at the point of escape for the gases can obviously have little or no influence on the furnace-flame temperature as a whole, even though it does reduce the gas temperature at the gas-discharging furnace boundary.

31 The conventions concerning the use of subscripts are noteworthy. Thus Q_{RC} denotes the radiation from the refractory to the cold surface, whereas Q_{CR}, now with the subscript letters reversed, indicates the heat absorbed at the cold surface by the total heat discharged to it from the refractory. The first letter in the subscript, i.e., C, thus indicates the place of energy measurement, and the second letter that from or toward which the energy discharge occurred. In some cases the second letter may be used

to qualify the quantity in other ways. Reflected energy is indicated by the symbol q, whence q_{RC} denotes the net reflection from the refractory to the cold surface, but q_{CR} denotes the net reflection at the cold surface of the total energy discharged to it from the refractory.

General Furnace Heat-Flow Equations

32 The time rate of energy loss from the flame, as shown by Equation [7], may be expressed in the form

$$dQ_F/dt = X_R+Q_{CR}+Q_{CT}+Z_C \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [8]$$

in which

$$Q_{CR} = (Q_{TR}+Z_R-Q_{AR}-X_R)(1-\beta_{CN}) \quad . \quad . \quad [9]$$

and

$$Q_{CT} = Q_{TC}\,(1-\beta_{CN}) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [10]$$

in which β_{CN} denotes the coefficient of reflection at the cold surface. Investigation shows that with an increase in furnace air temperature of 100 degrees, Q_{AR} represents approximately two per cent of the liberated energy. The heat Z_R transferred by *convection only*[1] to the interior refractory surface is likewise in the vicinity of one to three per cent of this energy. It follows that, within the limits of error involved, Z_R may be taken equal to Q_{AR}. With this assumption, Equation [8] may be expressed in the form

$$dQ_F/dt = (Q_{TR}+Q_{TC})(1-\beta_{CN})+Z_C \quad . \quad . \quad . \quad . \quad [11]$$

33 Since both the gaseous and solid fuel masses radiate, the above equation is now stated in the form

$$dQ_F/dt = (1-\beta_{CN})[(K_1'+K_1'')(T_U^4-T_R^4) + (K_2'+K_2'')(T_U^4-T_C^4)]+Z_C \quad . \quad . \quad . \quad [12]$$

in which the first product within the square brackets represents Q_{TR} and the second product Q_{TC}. Of the constants, K_1'' has such a value that the product $K_1''(T_U^4-T_R^4)$ represents the energy radiated to the refractory from the radiating gaseous portions and K_1' the corresponding value applicable between the solid fuel portions and refractory. The constants K_2' and K_2'' represent similar coefficients, applicable respectively between burned gaseous and solid fuel portions and the exposed boiler surfaces. Before Equation [12] may be solved the relation between the mean radiating refractory temperature T_R and the temperatures T_U and T_C obviously must be established.

34 An additional equation is furnished by the fact that the heat radiated from the refractory added to the heat loss through the refractory by conduction must be equal to the heat absorbed

[1] Differentiating between the heat transferred from the gases to the refractory by radiation and by convection.

by the refractory when the equilibrium state is reached as to the temperature T_R. In terms of the nomenclature of Fig. 8, this is expressed algebraically as

$$Q_{TR}+Z_R-Q_{AR}-X_R-q_{RC}=Q_{RC}$$

As before, Z_R, Q_{AR}, and X_R are very small quantities, whence the difference

$$Z_R-(Q_{AR}+X_R)$$

may be taken as zero, and

$$(Q_{TR}-q_{RC})=Q_{RC}$$

But q_{RC} is the net reflection from the refractory to the cold surface, whence the above equation is expressible as

$$Q_{TR}(1-\beta_{RN})=Q_{RC} \quad . \ . \ . \ . \ . \ . \ . \ . \ [13]$$

in which $(1-\beta_{RN})$ represents the net reflection coefficient from refractory to cold surface. In terms of the temperatures involved, this equation is then stated in the form

$$K_1(T_U^4-T_R^4)(1-\beta_{RN})=K_3(T_R^4-T_C^4) \ . \ . \ . \ [14]$$

in which $K_1=(K_1'+K_1'')$, and K_3 is a similar coefficient applicable between the refractory and cold surface. Solving [14]

$$T_R^4=\frac{(1-\beta_{RN})K_1T_U^4+K_3T_C^4}{(1-\beta_{RN})K_1+K_3} \ . \ . \ . \ . \ . \ . \ [15]$$

Placing $K_2'+K_2''=K_2$, and substituting [15] in [12], the resulting equation is

$$dQ_F/dt=(1-\beta_{CN})\left[\frac{K_1K_3}{(1-\beta_{RN})K_1+K_3}+K_2\right][T_U^4-T_C^4]+Z_C \ . \ . \ . \ [16]$$

35 This equation represents the furnace energy loss from flame and solid fuel portions. It differs from the heat absorption $(Q_{GR}+Q_{CT}+Z_C)$ at the exposed boiler surfaces by quantity (X_R+Q_{AR}), whence the heat absorption at the exposed surface is equal to

$$dQ_A/dt=dQ_F/dt-(X_R+Q_{AR}) \ . \ . \ . \ . \ . \ . \ [17]$$

36 By substituting from [2], [3], and [16] in [1], the equilibrium equation is stated in the final form

$$G_UU=\Sigma G_B\Delta h_B+(1-\beta_{CN})\left[\frac{K_1K_3}{(1-\beta_{RN})K_1+K_3}+K_2\right][T_U^4-T_C^4]+Z_C \ . \ . \ . \ [18]$$

For a given temperature T_C the magnitude of heat capacities Δh_B, as well as that of some of the constants K, depends, as will later be shown, on the temperature T_U. The solution for T_U is necessarily the result of a series of trials. The method is one of

assuming a trial value of T_U and for this evaluating all of the quantities dependent on it. Such values are substituted in [18] which is then solved for T_U. When the latter value agrees with the trial value on which the various magnitudes are based, the solution is correct in all details for the assumed furnace conditions. Before attempting a more exact definition of the various constants involved, it is now advantageous to examine the physical process in more detail.

Physical Nature of the Process

37 A particle such as a, Fig. 9, radiates both above and below the plane $p-p$. The plane $p-p$ cuts the particle into that portion whose surface can "see" through the solid angle 2π above the plane and that portion whose surface can "see" through the solid angle 2π below the plane. For symmetrical particles such as spheres, cubes, parallelograms, symmetrical crystal shapes, etc., each of the foregoing surfaces is one-half the total surface. Even for particles of other shapes the same assumption is permissible because as the particle travels through the furnace it will undoubtedly rotate about both the major and minor axes with a resulting time mean surface exposure in any given direction equal to one-half that of an envelope just large enough to contain the particle and tangent at the highest points. For convenience the particles may therefore be assumed to be small spheres, each of which has the same volume as that of the mean particle, whence the total radiation through a hemisphere (solid angle $= 2\pi$) is that from one-half the total spherical surface. Similarly the radiation to any point from the particle is the sum total of radiations to that point from all surface points which can "see" the receiving point. Any point can thus "be seen" by one-half of the surface of the particle. This becomes at once obvious if the plane $p-p$ cuts the particle in such a direction as to be perpendicular to a straight line connecting the point in question with the center of volume of the particle.

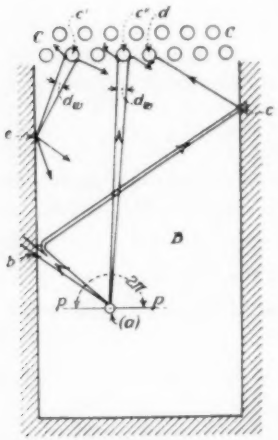

Fig. 9 Radiation from a Particle

38 Radiations from the particle are discharged in all directions. Some will strike the refractory R at points as b, others will strike the cold surface C, at points such as c''. A part of the radiations

striking at b are *reflected* as shown by the ray $b-c$. This *reflected* ray is largely absorbed at c whence only a small portion is finally reflected to the cold surface at d and a still smaller portion of the already reduced ray $d-c$ is reflected from the cold surface back into the furnace.

39 If the refractory furnace walls and the gaseous masses of the flame should be perfect reflectors and there should be no heating of such walls by convection, all radiations from the particle a would eventually be received at cold surface C with the same intensity as though discharged, without any intermediate interference, directly to the cold surface. If a is a black body under such conditions the heat absorption at C would be exactly equivalent to the condition in which the surface of a can "see" only the cold surface C. This corresponds to the assumption that the fuel surface is placed directly below the heat-absorbing surface and that the latter reflects none of the radiations striking it when measured according to the intensity $K(T_U^4-T_C^4)$. In a perfect reflecting furnace, therefore, the total absorption rate at the cold surface, equal to or greater than that of the fuel surface, would depend only on the fuel surface, its temperature, and the temperature of the heat-absorbing surface, the furnace height and other dimensions in no way reducing this in amount. Under such conditions the refractory furnace walls would tend to assume the temperature of the outside media with which they are in contact, and no heat loss by conduction through the walls could result.

40 Actually the reflective coefficients in practical furnaces are low, whence the refractory walls as mentioned absorb heat and assume a temperature T_R between that of the radiating body and the cold surface. Since T_R is greater than T_C, any point e of the refractory surface will radiate net energy to the cold surface. Radiation will also occur between the hotter and colder portions of the refractory.

41 Practically all of the radiated and reflected rays pass through the flame masses. A radiating surface thus "sees" the surface to which it is radiating as through a cloud. This cloud interferes with the ray unless the particles of the cloud are perfect reflectors, when the vision would be, as before mentioned, as though unobstructed. This flame or cloud interference is caused principally by the solid particles in suspension, but the CO_2 and H_2O molecules[1] likewise absorb an appreciable amount of the radiant energy which strikes them.

42 To study the reflection in more detail consider a particle such as o in the furnace of Fig. 10. The rays, a, b and c are reflected respectively one, two, and three times before striking the

[1] Wien, Julius, and others have shown that these are the only gaseous molecules which absorb and radiate appreciable amounts of heat.

cold surface. All rays striking between a and b are thus reflected once, whereas those between b and c, and c and o are reflected respectively two, and three or more times. The fraction of the energy radiated to the refractory which eventually is impinged on the boiler surfaces by the paths of reflection may thus be approximated by application of a weighted average such as

$$\beta_{NR} = \frac{\Delta a\beta_R + \Delta b\beta_R{}^2 + \Delta c\beta_R{}^3 + \ldots . \tfrac{1}{2}\Delta n\beta^n}{\Delta a + \Delta b + \Delta c + \ldots . \tfrac{1}{2}\Delta n} \quad . . [19]$$

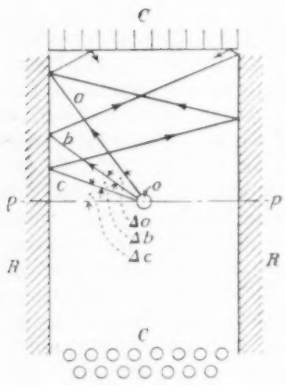

FIG. 10 STUDY OF REFLECTION

It is also apparent that the furnace shape and refractory distribution have some influence on the actual magnitude of β_{NR}. The fraction of total striking rays which are reflected only once is obviously seldom over one-third of the total striking rays, whence $\Delta\beta_{NR}$ will practically always be considerably less than the reflection coefficient β_R at a point of the refractory surface. If we may assume the angles $\Delta a = \Delta b = \Delta c$, $\Delta\beta_{NR}$ is very nearly $\frac{1}{2}\beta_R$, which value may be used as an approximation. The net reflection β_{NC} from the cold surface back into the flame is obviously equal to β_C, the coefficient of reflection of that surface when the surface is considered as one or more flat plates. For a tubular surface as shown at the bottom of Fig. 10, approximately one-half of the reflections would be directed deeper into the boiler, thus resulting in $\beta_{NC} = \frac{1}{2}\beta_c$.

CONSTANTS APPLICABLE BETWEEN SOLIDS AND SOLIDS FOR RADIATION THROUGH THE FLAME CLOUD

43 The radiation constant K_1' applicable between a solid particle in the flame and a refractory surface is now stated in the form

$$K'_1 = \alpha'_U I_g (1 - I_S) \frac{S'_U}{2} \times \frac{\omega'_{UR}}{2\pi} \quad . . . . . [20]$$

in which

α'_U = radiation coefficient at the fuel surface when radiating through a perfect reflecting or unclouded medium

I_g = radiation penetration coefficient through the gaseous media

I_S = mean shielding coefficient of solid particles for each other

ω'_{UR} = mean solid angle subtended by the receiving area from the radiating surface

and S'_U = total surface area of particles in suspension at any instant, sq. ft.

Similarly

$$K'_2 = \alpha'_U I_g (1-I_S) \frac{S'_U}{2} \times \frac{\omega'_{UC}}{2\pi} \quad . \quad . \quad . \quad . \quad . \quad [21]$$

in which ω'_{UC} represents the mean solid angle subtended by the cold surface from the radiating particles.

44 Evaluation of the solid angles ω'_{UR} and ω'_{UC} for receiving areas bounded by straight sides presents in general a problem in spherical trigonometry and the method of its application is outlined in Appendix No. 2. As a simple case, radiations to the faces of a cube from a particle at the center may be examined. Since all of the faces are equal and at right angles to each other there will be six equal solid angles whose sum must be equal to the solid angle of a sphere. Hence $6\omega = 4\pi$ and $\frac{\omega}{2\pi} = \frac{1}{3}$. In words, each face of the cube receives one-third of the radiations which a hemispherical shell would receive from the same particle surface if the plane base $p-p$ of the shell should cut the particle as shown in Fig. 9, the temperatures of the cube face and shell, of course, being of equal magnitude.

45 The refractory surfaces act as an extended solid fuel bed in radiating energy to all parts of the furnace. The net discharge is that through the solid angle subtended by the cold surface from the refractory. The form in which K_3 is expressed depends upon the place at which this discharged energy is to be analytically measured. Thus it is definable in terms of the discharge from the refractory wall toward the cold surface and also in terms of energy received at the cold surface by the discharge from the refractory wall. In terms of the first definition we have

$$K_3 = \alpha_{Rf} I_g (1-I_S) \frac{\omega_{RC}}{2\pi} S_R \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [22]$$

and in terms the second definition

$$K_3 = \alpha_{Rf} I_g (1-I_S) \frac{\omega_{CR}}{2\pi} S_C \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [23]$$

Radiation Intensity from Gaseous Masses

46 The net radiation exchange between a gaseous flame and a colder solid body is, as shown by Julius,[1] largely due to the CO_2 and H_2O molecules. Professor Tyndall[2] has shown that these

[1] Die Licht und Warmstrahlung verbrannten Gase.
[2] Preston, Theory of Heat.

products of combustion likewise absorb heat, which of course they should if they act as radiators. All other molecules in the gaseous mixture apparently act as nearly perfect reflectors.

47 Callendar found that the "intrinsic radiance" for a given kind of molecule depends on the density and thickness of the radiating gaseous column, but that when the flame depth is two feet or over in thickness, increased depth ceases to influence the radiation intensity per unit area of flame surface. A series of Meker burners utilizing Cambridge coal gas as a fuel were used in these experiments. Apparently the relation of (CO_2+H_2O) by weight in the products of combustion was about 22 per cent, and the radiation intensity based on the flame surface was 7 per cent that of a black body. The expected radiation intensities from the surface of any other flame should thus be related approximately as $\rho/0.22$, in which ρ represents the fraction by weight of (CO_2+H_2O) in the flame. For bituminous-coal flames the following values exist:

40 per cent excess air, $\rho = 0.23$
20 per cent excess air, $\rho = 0.22$

It thus appears reasonable to use a value of $\alpha''_U = 0.07\alpha_0$ for the radiation intensity at the flame surface. That is,

$$\alpha''_U = 0.07 \times 1.62 \times 10^{-9} = 1.135 \times 10^{-10} \quad . \; . \; . \quad [24]$$

It is interesting to note that practically all of the common fuels with the exception of blast-furnace gas yield a value $\alpha''_U = 0.07\alpha_0$. For blast-furnace gas $\rho = 0.32$, whence $\alpha''_U = 1.69 \times 10^{-10}$. The radiation coefficient of blast-furnace gas is thus relatively high, but the low flame temperature more than offsets this, with a resultant relatively low net heat transfer by radiation.

48 Since the radiation from the gaseous masses may be based wholly on their outside surfaces, the radiation constants K_1'' and K_2'' may be expressed within the limits of accuracy of other data by the relations

$$K''_1 = \alpha''_U \frac{\omega''_{UR}}{2\pi} S''_{UR} \quad . \; . \; . \; . \; . \; . \quad [25]$$

$$K''_2 = \alpha''_U \frac{\omega''_{UC}}{2\pi} S''_{UC} \quad . \; . \; . \; . \; . \; . \; . \quad [25a]$$

in which

S''_{UR} = flame surface which can "see" the refractory
S''_{UC} = flame surface which can "see" the cold surface
ω''_{UR} = mean solid angle subtended by refractory from flame surface which can "see" the refractory
ω''_{UC} = mean solid angle subtended by cold surface from flame surface which can "see" the cold surface.

The Coefficients α', I_g, I_s, and Other Quantities

49 The radiation coefficient α' is obtainable from the relation

$$\alpha' = (1-\beta)\alpha_0 \quad \ldots \ldots \ldots \quad [26]$$

in which β represents the reflection coefficient at the surface in question and α_0 the black-body radiation coefficient. α_0 is equal to 1.62×10^{-9} and β has the following approximate magnitudes:

Well-sooted metal surface, 0.05
Clean boiler tubes, unpolished, 0.10
Ordinary clean refractory, 0.10
Well-sooted refractory, 0.05.

50 The penetration coefficient I_g for radiations through the gaseous media may be approximated from the relation

$$I_g = \frac{\alpha_0 - \alpha'}{\alpha_0} \quad \ldots \ldots \ldots \quad [27]$$

This relation is further discussed in Appendix No. 3.

51 The value of the mean shielding coefficient I_S is obtainable from the relation[1]

$$I_S = 87.1\,\frac{V_f^{\frac{1}{3}}K_0^{\,2}}{G_B T_U \delta_0 d_0} - 64.2\times10^2\left[\frac{V_f^{\frac{1}{3}}K_0^{\,2}}{G_B T_U \delta_0 d_0}\right]^2 \quad [28]$$

in which

V_f = flame volume in cubic feet
G_B = pounds of burned gas per pound of fuel
T_U = mean radiating flame temperature, deg. fahr.
d_0 = initial mean diameter of the fuel particles
K_0 = coefficient such that $K_0 d_0$ is equal to the mean particle diameter with moisture and volatile expelled
δ_0 = weight of fuel per cubic foot.

The development of this relation is included in Appendix No. 4.

52 The total number N_t and area S'_U of particles in suspension is obtainable from the following relations:

$$N_t = 130.6\times10^3\,\frac{V_f}{G_B T_U \delta_0 d_0^{\,3}} \quad \ldots \ldots \quad [29]$$

$$S'_U = 950\,\frac{V_f K_0^{\,2}}{G_B T_U \delta_0 d_0} \text{ (sq. ft.)} \quad \ldots \ldots \quad [30]$$

The development of these expressions is included also in Appendix No. 4.

Black-Body Comparisons and Furnace-Absorption Efficiency

53 When a solution has been obtained which satisfies Equation [18], the results may be compared to the black-body intensities of the same temperatures. Such data are particularly valu-

[1] See limitations to applicability of Equation [28] in Appendix No. 4.

able when quick approximations of radiating conditions are to be attempted. In this respect the relative analytical measurements may advantageously be made in three ways.

54 For the first comparison the intensity of total radiation from the flame surface is compared to that of a black body having the same volume, surface, and temperature. The measurement is conveniently taken by considering the flame in question as completely enclosed by the cold surface C. The solid angles ω'_{VC} and ω''_{VC} now become equal to 4π. Since no refractory surface is now present, the surface S_R is obviously equal to zero. The ratio[1] of actual total radiation to that of the black-body flame is then expressed as

$$\frac{\alpha'_V I_g(1-I_S)S'_V+\alpha''_V S''_{VC}}{\alpha_0 S''_{VC}} = \mu_1 \quad . \quad . \quad . \quad . \quad [31]$$

The factor μ_1 is thus the relative radiance of the flame.

55 For the second comparison the furnace as a whole is compared to one in which the fuel surface is considered directly in front of the cold surface and equal to the latter in extent. For this condition the ratio[2]

$$\frac{\dfrac{K_1 K_3}{(1-\beta_{RN})K_1+K_3}+K_2}{\alpha_0 S_{VC}} = \mu_2 \quad . \quad . \quad . \quad . \quad . \quad [32]$$

yields a value whose magnitude measures the energy striking the cold surface by paths of radiation and net reflection as compared to that striking the cold surface by radiation from a black fuel bed immediately in front of it. It is really the radiating efficiency of the furnace as a whole.

56 A third comparison results from the ratio[3]

$$\frac{dQ_A/dt}{\alpha_0 S_{VC}(T_V^4-T_C^4)} = \mu_3 \quad . \quad . \quad . \quad . \quad . \quad . \quad [33]$$

This ratio measures the total heat absorption at the cold surface as compared to that striking this surface by the black fuel bed immediately in front of it. The quantity μ_3 is thus the relative absorption of the furnace as compared to that of a total black furnace with the same amount of cold surface.

57 The fourth coefficient μ_4 represents the furnace-absorption efficiency and is equal to the relation between the energy liberated and that absorbed by the exposed cold surface. This, however, does *not* include that absorbed as the result of convection by that cold surface at the aperture for discharging the gases into the convection zone of the heat-absorbing vessel. It does include that

[1] For development of this ratio see Appendix No. 5.
[2] See Appendix No. 5.
[3] See Equation [17].

absorbed by convection at other exposed cold surface. This quantity is evaluated from the relation [1]

$$\mu_4 = \frac{dQ_A/dt}{dQ_U/dt} \quad \ldots\ldots\ldots [34]$$

INFLUENCE OF PULVERIZATION FINENESS, FLAME VOLUME, ETC.

58 By application of Equation [31] with proper use of the expression [28] for the shielding coefficient, the variation of flame radiance with such conditions as volume V_f, combustion efficiency G_B, size of particle d_0, and temperature T_U may be ascertained. Some such results have been shown in the curves of Figs. 6 and 7 and need not be further discussed at this time.

59 When varying the particle size the limitations of Equation [28] should, as mentioned, be kept in mind, and for very small particles proper application of Equations [55] and [56] of Appendix No. 4 will yield values which are indicative of the trend of variation in the effective shielding.

60 These formulas are based on the inequality

$$(1-I_S)S'_U < S''_U \quad \ldots\ldots\ldots [34a]$$

in which S'_U represents the surface area of the suspended fuel particles, assumed as black bodies, and S''_U that of the flame envelope. If it is assumed that in the limit the relative radiance based on the radiation mean flame temperature T_U is unity, the inequality [34a] may be stated as an equation in which I_S is replaced with I'_S, the equivalent effective value of the mean shielding. The value of I'_S evaluated in this manner represents the minimum value which the mean shielding coefficient I_S on the visual basis (overlapping of surfaces) could have under such conditions. The real magnitude of I_S for indefinitely small particles is undoubtedly greater than I'_S. Under these conditions there must, even with a uniform particle size, uniformly distributed, and with a constant rate of energy liberation throughout, be a considerable slope to the temperature gradient from the flame center toward the outer surfaces. This is due to what might be called *series radiation* in which only the particles in a thin outer shell radiate directly to the furnace walls. The energy from the interior being transferred in a series of stages to particles in successive concentric flame shells whose thickness per shell depends upon the flame depth required for the complete visual shielding of any particle. The temperature gradient through the flame is thus dependent on the magnitude of the visual shielding. Both the decrease of particle size in a uniform mixture and the increase of congestion for a given size of particle thus increase the slope of this gradient. The foregoing analysis in Part I shows that in

[1] See Equations [17] and [2].

modern furnaces it is the congestion of particles of the sizes used, rather than the small size of the particle, which is a partial influence in causing the resulting temperature gradient.

61 As the particle diameter is increased indefinitely, the number N_f in suspension eventually becomes unity, in which case a single particle is equal to the fuel weight in suspension. The combustion rate is, obviously, seriously interfered with long before such a condition is reached, with a consequent reduction in flame temperature. A more complete analysis of the situation should therefore include the probable influence of pulverization fineness on the rate of energy liberation.

Oxygen Intensity

62 In discussing this condition it is convenient to introduce the factor mentioned in Part I of the paper and called oxygen intensity. For solid fuels it is, as defined, exactly the pounds of oxygen colliding per hour (and hence burning) per unit fuel-surface area. For gaseous fuels it is the pounds of oxygen per hour per cubic foot of gaseous mixture. If, then, u_0 represents the energy liberated per pound of oxygen uniting with the fuel in question, the products

$$u_0 \zeta_S S'_U = K'_U \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [35]$$

$$u_0 \zeta_V V_f = K''_U \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [36]$$

yield the total rates of energy liberation in B.t.u. per hour, in which

ζ_S = surface oxygen intensity, solid fuels

and

ζ_V = volume oxygen intensity, gaseous fuels.

63 The values of u_0 for various combustible constituents are given in the following table.

Combustible	u_0	Combustible	u_0
H_2	7600	C_2H_4	6390
C to CO_2	5440	C_2H_6	7240
C to CO	3250	Blast-furnace gas (avg.)	7620
CO to CO_2	7640	Coke-oven gas "	6700
CH_4, complete	6000	Natural gas "	6010
		Oil "	5720

The significant thing in this table appears as the relatively small deviation of calorific values when reduced to the per-pound-oxygen base.

64 The oxygen intensities ζ_S and ζ_V should not be confused with the rapidity of this process once a uniform mixture is established. For such cases the combustion is, of course, explosive in nature, and the rate is dependent on the character of the fuel constituents, temperature, and perhaps catalytic agents. The time required for combustion after a uniform mixture has been established is negligibly small compared to that required to create this mixture. The values of ζ_S and ζ_V with which this subject deals are thus really a measure of the speed of mixing.

65 We are concerned chiefly with the probable oxygen intensity at the surface of the fixed-carbon particles, but a brief discussion of such conditions as to the volatile portions is also essential. It is well known that for a given volume, pressure, and temperature all gases contain practically the same number of molecules, whence for a given initial distribution about the same time is required to create a uniform mixture with a given condition as to turbulence. In other words, it will take about the same time to distribute what oxygen is present throughout the mixture. The oxygen intensities and hence, by reference to the table in Par. 62, the rates of energy liberation, should therefore be nearly proportional to the quantities of oxygen contained per unit volume of the *burning gases in the furnace*. Investigation shows that such oxygen weights per unit volume differ by only small amounts, whence the possible energy-liberation rates for the very poor gases and the very rich gases should not differ very greatly. As a matter of fact they do not when the excess-air quantity is the same, although there may be a difference in flame length. A larger excess-air fraction, of course, reduces the degree of uniformity required as to mixture and hence the time for complete combustion. It therefore proportionately increases the probability of collisions between fuel and oxygen. Without going into more details in this respect it may now be said investigation shows, even when high-volatile coals are burned in the pulverized-fuel furnace, that the *required volume oxygen intensity ζ_V is far below that actually realized in the gas furnaces*. The burning of the volatile matter should therefore not at the present time limit the energy-liberation rate in pulverized-fuel furnaces.

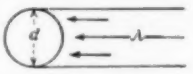

FIG. 11

66 Investigation of the oxygen intensities ζ_S at the surfaces of the particles, however, discloses another state of affairs. Let us examine the process shown in Fig. 11. Air is blown with a velocity λ toward the fuel sphere whose diameter is d. If the cross-sectional area of the stream is one square foot, the maximum volume in cubic feet per hour colliding is represented by the product 60λ, in which λ is the velocity in ft. per min. Then

$$\zeta_{S(\max.)} = \frac{60\lambda}{4.32RT_U/p} = 548\lambda/T_U \quad . \quad . \quad . \quad . \quad [37]$$

The following values result for ζ_S when $T_U = 3000$ deg. fahr.

λ	10	60	120
ζ_S	1.82	10.96	20.192

For an energy-liberation rate of 20,000 B.t.u. per cu. ft. per hr., which is near the limit with the most common methods of burning pulverized coal in boiler furnaces, the required ζ_S, with an initial

mean particle size of 0.002 in. diameter,[1] is approximately 10. It follows that the limiting mean relative velocity between the particles and the flame cannot be very much greater than 1 to 2 ft. per sec.

67 As the particles are decreased in size, the ratio of surface to weight per particle increases, whence the tendency will obviously be to reduce still further the relative velocities, and hence oxygen intensities, unless some other than the usual methods of creating turbulence can be found which is commercially satisfactory. If two flames should be directed one toward the other, the relative velocities would obviously be tremendously increased.

68 In the oil furnace a higher initial particle velocity is obtained by pressure, but this method does not lend itself to the pulverized-fuel system. Burners with perhaps a centrifugal discharging jet of high velocity may, when properly designed, offer a solution. With the present systems, however, it becomes apparent that finer pulverization *will probably not increase the rate* of energy liberation, because as the particle surface is increased the oxygen intensity ζ_g will probably decrease.

69 For modern furnaces then the curves of Figs. 6 and 7 are apparently as accurately representative as it is possible to predict.

PART III

Sample Calculations

70 The fuel used in the accompanying calculation is Illinois (Saline Co.) bituminous, of the following characteristics:

Proximate Analysis

	Per cent
Moisture	6.0
Volatile combustible	32.4
Fixed carbon	54.3
Ash	7.3

Ultimate Analysis

	Per cent
Carbon	71.6
Hydrogen	5.3
Oxygen	12.8
Sulphur	1.7
Nitrogen	1.3
Ash	7.3

Calorific value, B.t.u. per lb. 12,800

Weight of coal, lb. per cu. ft. ..... { 84 (raw) / 75 (coked)

Coal pulverized 75 per cent through 200-mesh screen
Mean initial diameter of particles, in. = 0.002 = d_0 (see App. No. 6)

71 The theoretical air required per pound of this fuel is

11.58 × 0.716 = 8.29 lb. for the carbon
34.8 × 0.037 = 1.29 lb. for the hydrogen
4.35 × 0.017 = 0.07 lb. for the sulphur

[1] See Appendix No. 6.

or a total of 9.65 lb. of air per lb. of fuel. Assuming 20 per cent excess air, the actual air required will be

$$9.65 \times 1.20 = 11.58 \text{ lb. per lb. of fuel}$$

The products of combustion of 1 lb. of the fuel will be as follows:

CO_2	=	0.716	×	44/12	=	2.625 lb.
H_2O	=	0.053	×	9	=	0.447 lb.
SO_2	=	0.017	×	2	=	0.034 lb.
Ash	=	0.073	×	1	=	0.073 lb.
N_2	=	(1.93 × 0.77)	+	(9.65 × 0.77) + 0.013	=	8.930 lb.
O_2	=	(1.93 × 0.23)	+	0	=	0.444 lb.
						12.583 lb.

11.58 lb. air + 1 lb. fuel = 12.58 lb. products.

72 Assuming that the particles are spherical, the volume of each particle entering the burner is $\frac{1}{6}\pi d_0^3$, where d_0 is the diameter of the particle.

73 The area of all the particles[1] in the flame at any instant is expressed by $\frac{1}{4}N\pi d_1^2$, where N is the number of particles in the flame, and d_1 is the average initial diameter of particles with moisture and volatile expelled. Moisture and volatile = 38.4 per cent of coal by weight. Then

$$d_1 = K_0 d_0$$

where K_0 is the coefficient of shrinkage. For coal weighing 84 lb. per cu. ft. raw and 75 lb. per cu. ft. coked,

$$K_0 = 0.884$$

and

$$d_1 = 0.884 d_0$$

74 The diameter of each particle entering the burner is $d_0 = 0.002$ in. and the volume is $\frac{1}{6}\pi d_0^3 = \frac{8}{6}\pi \times 10^{-9} = 4.19 \times 10^{-9}$ cu. in. The coal weighs 84 lb. per cu. ft. or 4.86×10^{-2} lb. per cu. in., and therefore each particle weighs $4.19 \times 10^{-9} \times 4.86 \times 10^{-2} = 2.04 \times 10^{-10}$ lb. The average surface area of each particle suspended in flame will be

$$a_s = \frac{1}{4}\pi d_1^2 = \frac{1}{4}\pi \times 3.133 \times 10^{-6} = 3.28 \times 10^{-6} \text{ sq. in.}$$

For a cubical furnace, 20 x 20 x 20 ft., the volume will be 8000 cu. ft. For the assumed conditions, the flame dimensions, 19 x 19 x 19 ft., will represent a volume of 6860 cu. ft. The flame surfaces parallel to exposed surfaces will be 361 sq. ft. each, the flame surfaces parallel to refractory surfaces will be 361 sq. ft. each, and the total flame surface will be 2166 sq. ft.

75 For an energy liberation of 15,000 B.t.u. per cu. ft. per hr., $dQ_U/dt = 15{,}000 \times 6860 = 102{,}900{,}000$ B.t.u. per hr.

[1] See Appendix No. 4.

Calculations for Case *D*

76 For Case *D* (see Par. 6) the exposed surface is 1600 sq. ft., distributed as follows: Two sides, bottom, top. (Aperture for escaping gases at top.) The exposed surface subject to convectional heat is 1200 sq. ft. and the refractory surface is 800 sq. ft.

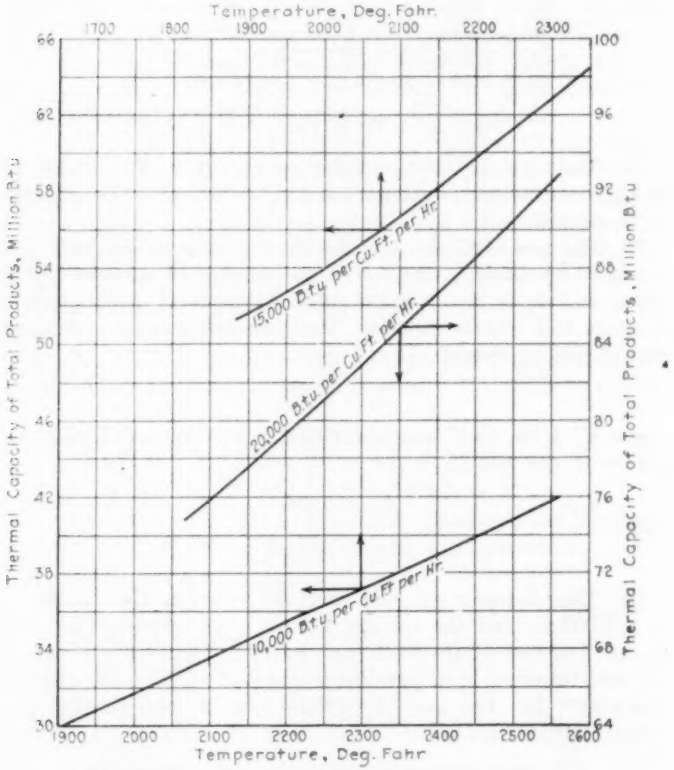

Fig. 12 Thermal-Capacity Curves, Total Products

77 The fuel required per hour will be $\frac{102{,}900{,}000}{12{,}800} = 8040$ lb., and the total weights of the products of combustion in lb. per hour will be as follows:

CO_2	$8040 \times 2.625 =$	21,160
N_2	$8040 \times 8.930 =$	71,850
O_2	$8040 \times 0.444 =$	3,570
SO_2	$8040 \times 0.034 =$	273
Ash	$8040 \times 0.073 =$	587
H_2O	$8040 \times 0.477 =$	3,840

Assuming $T_U = 2010$ deg. fahr. $= 2470$ deg. fahr. absolute, the total thermal capacity of products (from chart[1]) $= 54{,}400{,}000$

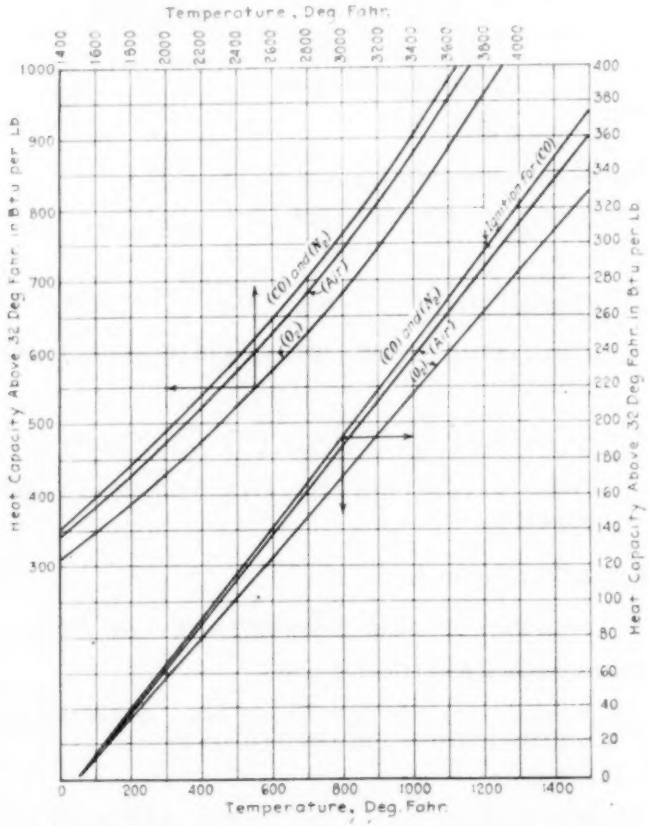

FIG. 13 THERMAL-CAPACITY CURVES, AIR, N_2, CO, AND O_2

B.t.u. per hr., and the weight of fuel present in flame at any instant will be

$$W_U = \frac{2116}{12.58 \times 53.3 \times 2470} \times 6860 = 8.75 \text{ lb.}$$

Then

$$N = \frac{8.75}{2.04 \times 10^{-10}} = 4.29 \times 10^{10}$$

[1] Sum of the products of weight of each constituent times its thermal capacity above 90 deg. fahr. obtained from thermal-capacity charts (App. No. 1). Resultant chart shown in Fig. 12.

and the shielding coefficient

$$I_S = 87.1\ \frac{V_f^{\frac{1}{3}}K_o^{\,2}}{G_B T_U \delta_0 d_0} - 64.2\times 10^2 \left[\frac{V_f^{\frac{1}{3}}K_o^{\,2}}{G_B T_U \delta_0 d_0}\right]^2$$

$$= \frac{87.1\times 19\times 0.7815}{12.58\times 2470\times 84\times 0.002} - 6420(0.00284)^2 = 0.20$$

and

$$1-I_S = 0.80$$

The effective surface area of particles is $\frac{1}{2}S'_U = \frac{1}{2}Na_s$

$$\tfrac{1}{2}S'_U = \frac{4.29\times 10^{10}\times 3.28\times 10^{-6}}{2\times 144} = 488 \text{ sq. ft.}$$

$$K_1 = \left(\alpha'_{Uf} I_g (1-I_S)\ \frac{S'_U}{2}\ \frac{\omega_{UR}}{2\pi}\right) n_1{}^* + \left(\alpha''_U\ \frac{\omega''_{UR}}{2\pi}\ S''_{UR}\right) n_1$$

$$= (1.62\times 10^{-9}\times 0.93\times 0.80\times 488\times \tfrac{1}{3})2 + (0.113\times 10^{-9}\times 1\times 361)2$$

$$= 480\times 10^{-9}$$

$$K_2 = \left(\alpha'_{Uf} I_g (1-I_S)\ \frac{S'_U}{2}\ \frac{\omega_{UC}}{2\pi}\right) n_2{}^\dagger + \left(\alpha''_U\ \frac{\omega_{UC}}{2\pi}\ S''_{UC}\right) n_2$$

$$= (1.62\times 10^{-9}\times 0.93\times 0.80\times 488\times \tfrac{1}{3})4 + (0.113\times 10^{-9}\times 1\times 361)4$$

$$= 960\times 10^{-9}$$

$$K_3 = \alpha_0 (1-\beta_R) I_g (1-I_S)\ \frac{\omega_{RC}}{2\pi}\ S_R$$

$$= 1.62\times 10^{-9}\times 0.93\times 0.93\times 0.80\times 0.875\times 800 = 800\times 10^{-9}$$

The convectional heat to cold surface

$$Z_C = S_Z U D; \quad U^{\ddagger} = 4; \quad D = 700 \text{ deg. fahr.}; \quad S_Z = 1200 \text{ sq. ft.}$$

$$= 1200\times 4\times 700 = 3{,}360{,}000 \text{ B.t.u. per hr.}$$

78 Then

$$dQ_F/dt = (1-\beta_{NC})\left[\frac{K_1 K_3}{(1-\beta_{NR})K_1 + K_3} + K_2\right][T_U{}^4 - T_C{}^4] + Z_C$$

$$= 0.97\left[\frac{480\times 10^{-9}\times 800\times 10^{-9}}{\{(0.97\times 480)+800\}\times 10^{-9}} + (960\times 10^{-9})\right]$$

$$[T_U{}^4 - T_C{}^4] + 3{,}360{,}000$$

The temperature of the exposed surfaces is 900 deg. fahr. abs. and $T_C{}^4 = 656\times 10^9$.

$$dQ_F/dt = (1225\times 10^{-9}\times T_U{}^4) + 2{,}560{,}000$$

$$dQ_U/dt = dQ_F/dt + dQ_B/dt$$

* n_1 denotes number of flame faces presented to refractory.
† n_2 denotes number of flame faces presented to cold surface.
‡ U denotes coefficient of heat transfer by convection.

$$102{,}900{,}000 = (1225 \times 10^{-9} \times T_U^4) + 2{,}560{,}000 + 54{,}400{,}000$$

$$T_U^4 = \frac{46}{1.225} \times 10^{12} = 37.5 \times 10^{12}$$

$$T_U = \sqrt[4]{37.5 \times 1000} = 2475 \text{ deg. fahr., abs.} = 2015 \text{ deg. fahr.}$$

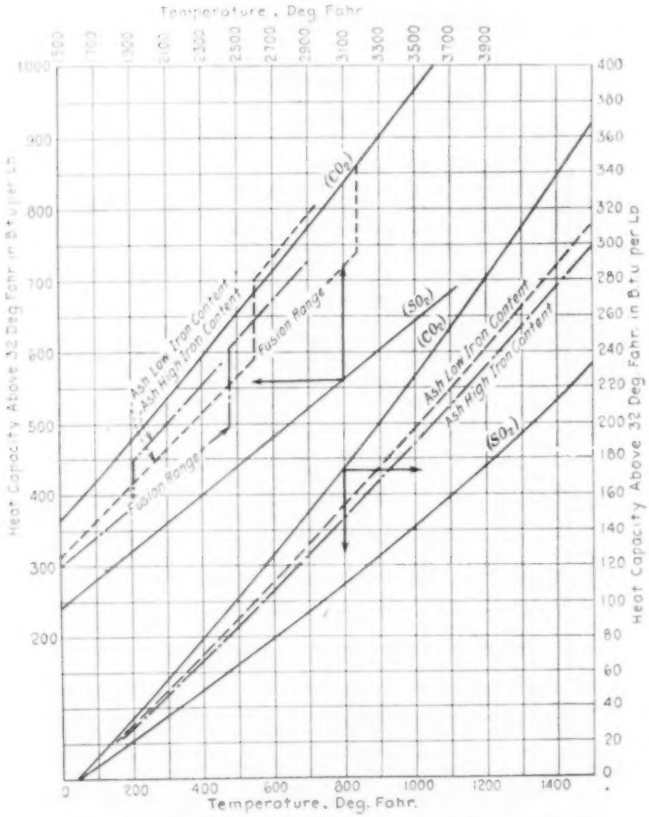

FIG. 14 THERMAL-CAPACITY CURVES, CO_2, SO_2, AND ASH

which checks with the initial assumption of 2010 deg. fahr.

$$T_R^4 = \frac{(1-\beta_{NR})K_1 T_U^4 + K_3 T_C^4}{(1-\beta_{NR})K_1 + K_3}$$

$$= \frac{(0.97 \times 480 \times 10^{-9} \times 37.5 \times 10^{12}) + (800 \times 10^{-9} \times 656 \times 10^9)}{(0.97 \times 480 \times 10^{-9}) + (800 \times 10^{-9})}$$

$$= 14.35 \times 10^{12}$$

$$T_R = \sqrt[4]{14.35 \times 1000} = 1945 \text{ deg. fahr., abs.} = 1485 \text{ deg. fahr.}$$

$$\mu_1 = \frac{\alpha'_V I_g (1-I_8) S'_V + \alpha''_V S''_{VC}}{\alpha_0 S''_{VC}}$$

$$= \frac{(0.93 \times 0.814 \times 976) + (0.07 \times 2166)}{2166} = 0.411$$

$$\mu_2 = \frac{\dfrac{K_1 K_3}{(1-\beta_{NR})K_1 + K_3} + K_2}{\alpha_0 S_{UC}} = \frac{1263 \times 10^{-9}}{2590 \times 10^{-9}} = 0.488$$

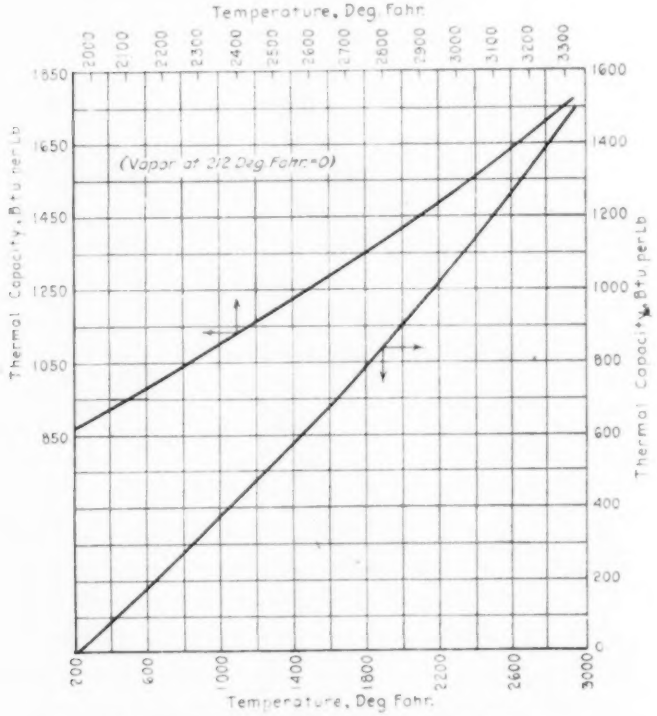

FIG. 15 THERMAL CAPACITY OF WATER VAPOR

$X_R = 1{,}500{,}000$ B.t.u. per hr. (heat losses through walls)

$$dQ_A/dt = dQ_F/dt - X_R = 48{,}500{,}000 - 1{,}500{,}000$$

$$= 47{,}000{,}000 \text{ B.t.u. per hr.}$$

$$Q_A = \frac{47{,}000{,}000}{1600} = 29{,}375 \text{ B.t.u. per sq. ft. per hr.}$$

$$\mu_3 = \frac{Q_A}{\alpha_0 (T_V^4 - T_C^4)} = \frac{29{,}375}{59{,}600} = 0.493$$

$$\mu_4 = \frac{dQ_A/dt}{dQ_U/dt} = \frac{47{,}000{,}000}{102{,}900{,}000} = 0.456$$

APPENDIX NO. 1

79 In computing the data by means of which the heat-capacity curves are plotted the following procedure was employed.

$$\left[H_p\right]_{T_1}^{T_2} = \int_{T_1}^{T_2} Y_p dT \qquad \text{and} \quad \left[h_p\right]_{T_1}^{T_2} = \int_{T_1}^{T_2} C_p dT$$

in which

Y_p = instantaneous molecular specific heat at constant pressure
C_p = instantaneous specific heat per pound at constant pressure
H_p = heat capacity, B.t.u. per mol
h_p = heat capacity, B.t.u. per pound.

$$C_p = \frac{Y_p}{\text{molecular weight}} \qquad \text{and } h_p = \frac{H_p}{\text{molecular weight}}$$

80 With two exceptions the specific-heat formulas used are based on the collected data and developed curves of Goodenough and Felbeck.[1] The data there included are taken from the best available sources. For sulphur dioxide and ash, other material is used, for which further details are included below. The specific-heat equations are now stated.

81 *Diatomic Gases* CO, N_2, *and* O_2.

$$Y_p = 6.93 + 0.1200 \times 10^{-3} T^2$$

calculated by Goodenough and Felbeck to fit data from measurements of Swann, Pier, and Bjerrum.

82 *Air.* Computed as a weighted mean from above values of N_2 and O_2.

83 *Carbon Dioxide,* CO_2

$$Y_p = 6.4587 + 5.0668 \times 10^{-3} T - 1.2480 \times 10^{-6} T^2 + 0.1086 \times 10^{-9} T^3$$

Pier's equation modified by Goodenough and Felbeck to fit data from measurements of Swann, Pier, and Bjerrum.

84 *Sulphur Dioxide,* SO_2. Computed from CO_2 values by assuming the same molecular specific heat for both substances. This method probably yields better results than are available from present experimental data of this substance.

85 *Water Vapor*

$$Y_p = 8.33 - 0.276 \times 10^{-3} T + 0.423 \times 10^{-6} T^2$$

Calculated by Goodenough and Felbeck to fit experimental data of Pier and Bjerrum at the high temperatures and those of Knoblauch and Jakob and Holborn and Henning at the low temperatures.

86 *Ash.* Any curves for ash can necessarily be only approximately correct due to the variability of ash and also because of the meager data concerning the properties of many of its constituents.

Heat-Capacity Curves

87 With the exception of water vapor these curves, Figs. 12-15, have been plotted to show the heat capacity above 32 deg. fahr. For the sake of convenience in calculation, that of water vapor, Fig. 15, is plotted with 212 deg. fahr. as the zero point. Still further to facilitate the computations, heat-capacity curves based on these values have been plotted for the actual burned mixture. These are shown with the sample calculations, Part III.

[1] Bulletin No. 139, Engineering Experiment Station, Univ. of Ill.

APPENDIX NO. 2

88 In Fig. 16, α = angle aob, β = angle boc and γ = angle aoc. $s = \frac{1}{2}(\alpha+\beta+\gamma)$.

89 ***The Cube.*** In this case

$$A = B = C$$

and $$\omega_{AB} = \omega_{AC} = \omega_{BC}$$

(*a*) $$6\omega = 4\pi, \text{ whence } \omega/2\pi = 0.333$$

(*b*) By L'Huilier's formula

$$\omega/2\pi = \frac{4 \arctan\sqrt{\tan\tfrac{1}{2}s \tan\tfrac{1}{2}(s-\alpha) \tan\tfrac{1}{2}(s-\beta) \tan\tfrac{1}{2}(s-\gamma)}}{180}$$

$$r = \tfrac{1}{2}\sqrt{A^2+B^2+C^2} = \tfrac{1}{2}A\sqrt{3}; \qquad \sin\frac{\alpha}{2} = \frac{A}{2r} = \frac{1}{\sqrt{3}} = 0.5774;$$

$$\sin\frac{\beta}{2} = \frac{B}{2r} = \frac{1}{\sqrt{3}}; \qquad \sin\frac{\gamma}{2} = \frac{\sqrt{B^2+C^2}}{2r} = \frac{\sqrt{2}}{\sqrt{3}} = 0.8164$$

$$\frac{\alpha}{2} = 35°27 \qquad \frac{\beta}{2} = 35°27 \qquad \frac{\gamma}{2} = 54°73 \qquad s = 125°27$$

$$\alpha = 70°54 \qquad \beta = 70°54 \qquad \gamma = 109°46$$

$$\begin{aligned} \tan\tfrac{1}{2}s &= \tan 62°63 = 1.932 \\ \tan\tfrac{1}{2}(s-\alpha) &= \tan 27°37 = 0.5177 \\ \tan\tfrac{1}{2}(s-\beta) &= \tan 27°37 = 0.5177 \\ \tan\tfrac{1}{2}(s-\gamma) &= \tan\ 7°90 = 0.1388 \end{aligned}$$

$$\sqrt{1.932\times0.5177^2\times0.1388} = 0.5177\sqrt{0.268} = 0.5177\times0.518 = 0.2682$$

$$\arctan 0.2682 = 15°0$$

$$\frac{\omega}{2\pi} = \frac{4\times15}{180} = 0.333$$

90 ***Special Parallelopiped.*** In this case let $A = B = 2C$. If C is unity, $A = B = 2$.

(*a*) $$\omega_{AB}+\omega_{BC}+\omega_{AC} = 2\pi$$

(*b*) L'Huilier's formula

$$r = \tfrac{1}{2}\sqrt{A^2+B^2+C^2} = \frac{3c}{2}$$

$$\sin\frac{\alpha}{2} = \frac{A}{2r} = \frac{2c}{2r} = 0.6667 \qquad \frac{\alpha}{2} = 41°81$$

$$\sin\frac{\beta}{2} = \frac{B}{2r} = \frac{2c}{2r} = 0.6667 \qquad \frac{\beta}{2} = 41°81$$

$$\sin\frac{\gamma}{2} = \frac{\sqrt{A^2+B^2}}{2r} = \frac{2C\sqrt{2}}{2r} = 0.943 \qquad \frac{\gamma}{2} = 70°58$$

$$\alpha = 93°62 \qquad \beta = 93°67 \qquad \gamma = 141°16$$

$$\begin{aligned} \tan\tfrac{1}{2}s &= \tan 77°10 = 4.366 \\ \tan\tfrac{1}{2}(s-\alpha) &= \tan 30°29 = 0.5841 \\ \tan\tfrac{1}{2}(s-\beta) &= \tan 30°29 = 0.5841 \\ \tan\tfrac{1}{2}(s-\gamma) &= \tan\ 6°52 = 0.1143 \end{aligned}$$

$$\sqrt{4.366\times0.5841^2\times0.1143} = 0.5841\sqrt{0.499} = 0.4130$$

$$\arctan 0.413 = 22°43$$

$$\omega/2\pi = \frac{4\times22.43}{180} = 0.50$$

$$\sin\omega_{AB} = \pi$$

and $$\omega_{AB}+\omega_{AC}+\omega_{BC} = 2\pi$$

$$\omega_{AC}+\omega_{BC} = \pi$$

By symmetry $\omega_{AC} = \omega_{BC}$

$$\therefore \omega_{AB} = \pi, \quad \frac{\omega_{AB}}{2\pi} = 0.500$$

$$\omega_{BC} = \pi/2, \quad \frac{\omega_{BC}}{2\pi} = 0.250$$

$$\omega_{AC} = \pi/2, \quad \frac{\omega_{AC}}{2\pi} = 0.250$$

91 *Special Parallelopiped.* In this case let $A = 2B = 2C$. If $B = C$ is unity; $A = 2$.

(1) $\omega_{AB} = \omega_{AC}$ by symmetry

(2) $\omega_{AB}+\omega_{AC}+\omega_{BC} = 2\pi$

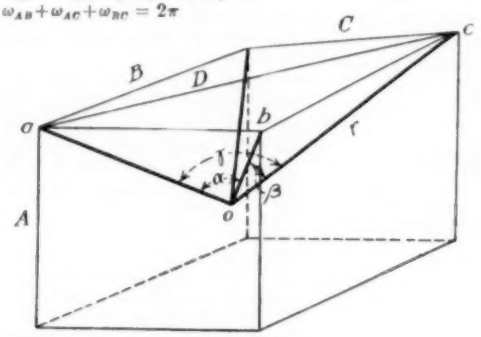

FIG. 16

(*a*) L'Huilier's formula for ω_{BC}

$r = \frac{1}{2}\sqrt{A^2+B^2+C^2} = \frac{1}{2}C\sqrt{6}$

$$\sin\frac{\alpha}{2} = \frac{C}{2r} = \frac{1}{\sqrt{6}} = \frac{1}{2.449}; \quad \frac{\alpha}{2} = 24°.10, \quad \alpha = 48°.20$$

$$\sin\frac{\beta}{2} = \frac{B}{2r} = \frac{1}{\sqrt{6}} = \frac{1}{2.449}; \quad \frac{\beta}{2} = 24°.10, \quad \beta = 48°.20$$

$$\sin\frac{\gamma}{2} = \frac{\sqrt{B^2+C^2}}{2r} = \frac{\sqrt{2}}{\sqrt{6}} = \frac{1}{\sqrt{3}}; \quad \frac{\gamma}{2} = 35°.27, \quad \gamma = 70°.54$$

$s = \frac{1}{2}(\alpha+\beta+\gamma) = 83°.47$

$$\tan\tfrac{1}{2}s = \tan 41°.73 = 0.892$$
$$\tan\tfrac{1}{2}(s-\alpha) = \tan 17°.63 = 0.3178$$
$$\tan\tfrac{1}{2}(s-\beta) = \tan 17°.63 = 0.3178$$
$$\tan\tfrac{1}{2}(s-\gamma) = \tan 6°.47 = 0.1134$$

$$\sqrt{0.892\times 0.3178^2\times 0.1134} = 0.3178\sqrt{0.101} = 0.101$$

$$\text{arc tan } 0.101 = 5°.7$$

$$\omega/2\pi = \frac{4\times 5.7}{180} = 0.125$$

$$\omega_{BC} = \tfrac{1}{4}\pi$$

$$\omega_{AB} = \omega_{AC} = \tfrac{7}{8}\pi$$

$$\omega_{AB} = \tfrac{7}{8}\pi, \quad \frac{\omega_{AB}}{2\pi} = 0.4375$$

$$\omega_{AC} = \tfrac{7}{8}\pi, \quad \frac{\omega_{AC}}{2\pi} = 0.4375$$

$$\omega_{BC} = \tfrac{1}{4}\pi, \quad \frac{\omega_{BC}}{2\pi} = 0.1250$$

APPENDIX NO. 3

COEFFICIENT OF PENETRATION, I_g

92 Referring to Fig. 17, consider a fuel bed S'_v which radiates to the cold surface C through the gaseous medium, containing some radiating molecules. Suppose that the intrinsic radiance for the gas alone through the surface $\Delta S''_v$ is a''_v and for the fuel bed unshielded is a'_v. It is apparent that the maximum value of the intrinsic radiance through the surface $\Delta S''_v$ can be no greater than that of a black body for which $a = a_0$. If then S'_v is considered as a black body its radiation must be interfered with by the gas molecular radiance to such an extent that the total radiations through $\Delta S''_v$ never exceed the intensity $a = a_0$.

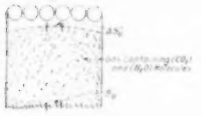

FIG. 17

93 To correct the radiation intensity from the fuel bed through the surface $\Delta S''_v$, two equation forms may be considered. Letting a'_{vN} represent the net radiance from the fuel bed through the surface, these equations may be expressed as

$$a'_{vN} = a'_v - a''_v$$

or

$$a'_{vN} = \frac{a_0 - a''_v}{a_0} a'_v$$

Either form results in the same magnitude at the limits, but the first reduces to zero when $a'_v = a''_v$, regardless of the value of a'_v. The second reduces to zero only when a''_v is equal to a_0. Both expressions also yield the same result for any value of a''_v if a'_v is equal to a_0. Since the latter value of a'_v is closely approached in the furnace it makes practically no difference which form is adopted concerning radiations from the fuel bed.

94 For radiations from a highly reflective refractory there will obviously be a difference in this respect, and should a''_v be equal to a_R, radiant heat could be transferred from the refractory to the cold surface (according to the first form) only if the flame temperature were lower than that of the refractory, forcing thereby radiations from the refractory to strike the flame molecules and then be redirected to the cold surface.

95 Such a condition appears contrary to the observed facts. Furthermore if two thin gaseous strata of equal surface radiances are placed one back of the other, the total radiance through the outer surface is materially greater than that of either gas stratum by itself, whereas by the first form of the equation it should not increase in value. The second form is therefore adopted whence the coefficient of penetration is expressed by the relation

$$I_g = \frac{a_0 - a''_v}{a_0}$$

APPENDIX NO. 4

SHIELDING COEFFICIENTS I_s, SURFACE S'_v, ETC.

96 As a means of approximating the mean shielding between par-

ticles the following formulas are developed. Referring to Fig. 18, let

ΔF = cross-sectional area of the mean particles
x_m = mean distance between particles in the furnace
N_x = number of particles interfering with the radiation of a particle A.

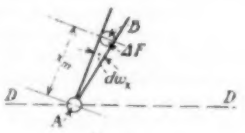

FIG. 18

97 The particle B will interfere with radiations of particle A to the extent of the solid angle $d\omega_x$ cut out by B. This may be measured in terms of the total radiation from the particle above or below the plane $D-D$. The interference fraction per particle B with reference to particle A is obviously $d\omega_x/2\pi$. For small particles this is equal to

$$\frac{\Delta F}{2\pi x_m^2} \quad \ldots\ldots\ [38]$$

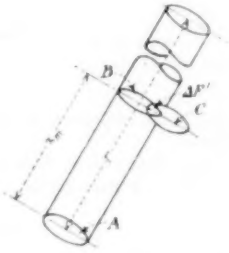

FIG. 19

98 There may, however, be other particles projecting partially or wholly into the cylindrical tube containing B and A. The shielding by such a particle as B, of A, is then only partial because some of the shielding surfaces overlap. Considering all the particles in a given tube as at the mean distance x_m from the particle A, this overlapping appears as shown in Fig. 19.

FIG. 20

99 Considering further a uniform particle distribution, the probability is such that there will be on the average

$$P = \Delta F r (N_t/V_f) \quad \ldots\ldots\ [39]$$

other particles within the tube of length r, in which P may be anything from zero to infinity, depending upon the number of particles N_t/V_f per unit of flame volume.

100 The net shielding area of the two particles B and C is now equal to $(2\Delta F - \Delta F')$, whence per particle the net shielding area reduces to

$$\Delta F - \tfrac{1}{2}\Delta F' \quad \ldots\ldots\ [40]$$

This is true for cases in which P is equal to or less than unity.

101 But $\Delta F' = P\Delta F$, whence the net shielding area per particle is expressed in the form

$$(1-\tfrac{1}{2}P)\Delta F \quad \ldots\ldots\ [41]$$

102 The total number of particles N_x which are in a position to interfere with the particle surface above the plane $D-D$ is equal to $\frac{1}{2}N_t$, whence the shielding coefficient I_s may be expressed as

$$I_s = (1-\tfrac{1}{2}P)\frac{\Delta F}{2\pi x_m^2} \times \tfrac{1}{2}N_t \quad \ldots\ldots\ [42]$$

Before this equation may be used certain quantities will have to be evaluated in terms of the furnace conditions.

103 *Mean Distance* x_m. Considering a particle A (Fig. 20) as surrounded by particles in concentric spheres of distance Δx apart and with K_x particles on the surface of the inner sphere, the mean distance x to any particle may obviously be computed by means of a weighted average. The number of particles in successive spherical surfaces is proportional to the area of such surfaces. The respective distances may thus be weighted by these areas. Hence any surface of radius x has the weight $4\pi x^2$ and the mean value of x is obtained from the relation

$$x_m = \frac{\int_{x=0}^{x=r} 4\pi x^2 \cdot x\,dx}{\int_{x=0}^{x=r} 4\pi x^2 dx} = \tfrac{3}{4}r \quad \ldots\ldots\ldots [43]$$

The mean distance is thus three-fourths the distance to the outer sphere.

104 *Equivalent Radius* r_e *for Volume Shapes Other Than Spheres.* Let V_f = flame volume. Then r_e may be found from the relation

$$\tfrac{4}{3}\pi r_e^3 = V_f$$

and

$$r_e = \sqrt[3]{\frac{3V_f}{4\pi}}$$

and

$$x_m = \tfrac{3}{4}r_e = \tfrac{3}{4}\sqrt[3]{\frac{3V_f}{4\pi}} \quad \ldots\ldots\ldots [44]$$

105 *Fuel Quantities in Suspension.* The fuel weight in suspension may be approximated with sufficient accuracy by applying the characteristic equation of a perfect gas with the air constant. Thus the gas weight in the furnace is expressed as

$$W_g = \frac{P_f V_f}{RT_v}$$

Fuel weight W_v in suspension is W_g/G_B, in which G_B represents the pounds of gas per pound of fuel. The resulting equation is

$$W_v = 39.45\,\frac{V_f}{G_B T_v} \quad \ldots\ldots\ldots [45]$$

106 *The Total Number of Particles in Suspension* may then be expressed as

$$N_t = 1.728\times 10^3\,\frac{W_v}{\delta_v \sigma_v}$$

in which δ_v = weight per cu. ft. of fuel and σ_v = initial mean volume per particle. For spherical particles, then,

$$N_t = 130.6\times 10^3\,\frac{V_f}{G_B T_v \delta_v d_v^3} \quad \ldots\ldots\ldots [46]$$

and

$$N_t/V_f = 130.6\times 10^3\,\frac{1}{G_B T_v \delta_v d_v^3} \quad \ldots\ldots\ldots [46a]$$

107 *The Time Mean of the Surface Area* is found from the relation

$$S'_v = \frac{N_t \int_{r=r_1}^{r=0} A\,dt}{\int_{r=r_1}^{r=0} dt} = \frac{N_t \int_1^2 4\pi r^2 dt}{\int_1^2 dt} \quad \ldots\ldots [47]$$

Now $$d\sigma = N_t 4\pi r^2 dr$$

also $$d\sigma = N_t \frac{\zeta_s}{2.67\delta_1} 4\pi r^2 dt$$

in which ζ_s represents the surface oxygen intensity[1] in lb. per sq. ft. fuel surface per hr., δ_1 represents the fuel density in lb. per cu. ft., and r the radius of the particle in feet. Hence

$$dt = \frac{2.67\delta_1}{\zeta_s} dr \quad \ldots\ldots [48]$$

Substituting in [47]

$$S'_v = \frac{N_t 4\pi \int_{r=r_1}^{r=0} \frac{r^2}{\zeta_s} dr}{\int_{r=1}^{r=0} \frac{dr}{\zeta_s}} \quad \ldots\ldots [49]$$

Considering the oxygen intensity as constant, this integrated value is finally expressed as

$$S'_v = N_t \frac{\pi}{3} d_i^2 = N_t \frac{\pi}{3} (K_o d_o)^2 \quad \ldots\ldots [50]$$

By a similar method the mean cross-sectional area ΔF is expressed as

$$\Delta F = \frac{\pi}{1728} (K_o d_o)^2 \text{ sq. ft.} \quad \ldots\ldots [51]$$

in which d_o is expressed in inches and d_i is the initial mean diameter of the particle with volatile and moisture expelled, which, as stated in Part III, is used because of the probable early expulsion of these constituents.

108 By substituting [46] in [50], the expression for the integrated value of S'_v finally appears in the form

$$S'_v = 950 \left[\frac{V_f K_o^2}{G_B T_U \delta_o d_o} \right] \text{(sq. ft.)} \quad \ldots\ldots [52]$$

109 Finally, substitution of [44], [46], and [51] in [42] yields the equation

$$I_s = 87.1 K_4 - 64.2 \times 10^3 K_4^2 \quad \ldots\ldots [53]$$

in which

$$K_4 = \frac{V_f \frac{1}{3} K_o^2}{G_B T_U \delta_o d_o} \quad \ldots\ldots [54]$$

110 This equation applies with fair accuracy when the mean particle size is such that the initial mean diameter d_o is not less than 1×10^{-3} in. The curve for variation of I_s, and hence μ_1 as well, with d_o may be plotted for the range of particles larger than $d_o = 1 \times 10^{-3}$ in. by means

[1] See section on influence of pulverization fineness, Pars. 58-60.

of data computed from this relation. A knowledge of the limit which μ_1 must approach will then make possible further approximations. Thus the limiting value of μ_1 is of course unity. For exceedingly small particles the total fuel-surface area becomes very large. In fact, it may eventually be many times the area of the flame-surface envelope. It follows that as the particle surface area S'_v assumes a value several times that of the total flame envelope S_v'', the minimum value of the visual shielding coefficient is such that

$$1 - I'_s = S''_v / S'_v \quad \ldots\ldots\ldots\ldots \quad [55]$$

in which S'_v may be computed from Equation [52]. Hence for values of S'_v several times greater than S_v'' a solution of the equation

$$1 - I'_s = \frac{S''_t G_B T_v \delta_v d_v}{950 V_f K_a^2} \quad \ldots\ldots\ldots\ldots \quad [56]$$

yields a value of I'_s which is indicative of the trend of variation in the effective shielding. A more detailed explanation of this phase is included in Par. 59.

APPENDIX NO. 5

BLACK-BODY COMPARISONS

111 The relative radiance at the flame surface is conveniently found by considering the flame completely enclosed in a cold black envelope, whence $\omega_{vc} = 4\pi$, and the total radiation is expressed by

$$\left[\alpha'_t I_g (1 - I_s) \frac{S'_v}{2} \left(\frac{4\pi}{2\pi} \right) + \alpha''_t S''_{vc} \right] [T_v^4 - T_c^4]$$

and a perfectly black flame would radiate at the rate

$$\alpha_v S''_{vc} [T_v^4 - T_c^4]$$

in which S''_{vc} now represents the exterior flame area. It follows that for the same values of T_v and T_c in both cases the relative radiance is expressed by the relation

$$\mu_1 = \frac{\alpha''_v I_g [1 - I_s] S'_v + \alpha''_t S''_{vc}}{\alpha_v S''_{vc}} \quad \ldots\ldots\ldots \quad [57]$$

112 The radiating efficiency of a furnace as defined in Part III of the paper is deduced in the same manner by relating the total energy actually striking the cold surface by virtue of radiation and reflection to that which would strike it with a black fuel bed immediately in front of it; the actual cold surface being now considered.

113 From Equation [16] it becomes obvious that the former is

$$\left[\frac{K_1 K_3}{(1 - \beta_{RN}) K_1 + K_3} + K_2 \right] (T_v^4 - T_c^4)$$

The latter is obviously equal to

$$\alpha_v S_{vc} (T_v^4 - T_c^4)$$

whence

$$\mu_2 = \frac{\left[\frac{K_1 K_3}{(1 - \beta_{RN}) K_1 + K_3} + K_2 \right]}{\alpha_v S_{vc}} \quad \ldots\ldots\ldots \quad [58]$$

114 The relative absorption term μ_3 is as shown based on the actual heat absorption at the cold surface and needs no further comment.

115 The furnace absorption efficiency μ_4 might be based either on the heat reclaimed by the incoming air plus that absorbed by the cold surface or on only that absorbed at the cold surface. In the former case the denominator should include the total energy supplied to the furnace Q_T, whereas in the latter case only the liberated energy is there included. Either form will thus yield practically the same value of μ_4. The latter form is therefore used because it better expresses the quantity in which the boiler designer is interested. It is to be again noted that the absorption by convectional heat Z_C by the cold surface at the gas aperture is not included in the numerator and should thus enter into any calculations for heat absorption in the principal convection zone of the boiler.

APPENDIX NO. 6

INITIAL MEAN DIAMETER OF PARTICLES, d_i (FIG. 21)

$$d_c = \frac{1}{(\text{mesh})}$$

Hence $$d_p \text{ (max.)} = (d_c - d_w) = \frac{1}{(\text{mesh})} - d_w$$

Surface area per lb. $\propto \frac{1}{\text{diameter of particle}}$

Total surface per lb. $\propto \sum w_i \frac{1}{d_i}$

Surface area per lb. $\propto w \frac{1}{d}$

d_i = mean initial diameter of particles

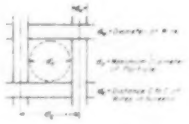

FIG. 21

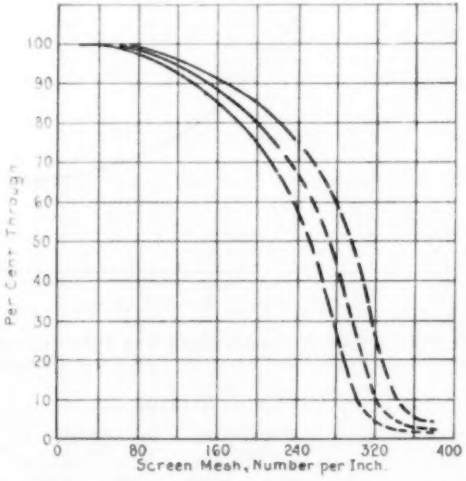

FIG. 22 PARTICLE DISTRIBUTION AS GROUND AND MEASURED

Hence

$$d_0 = \frac{1}{w_1/d_1 + w_2/d_2 + \ \ldots \ w_n/d_n}$$

TABLE 2 FOR BUREAU OF MINES STANDARDS

Mesh	1 Mesh	d_w In.	d_p(max.) In.	d_p(used) In.
48	0.0209	0.0092	0.0117	0.0110
65	0.0154	0.0072	0.0082	0.0075
100	0.0100	0.0042	0.0058	0.0054
150	0.0067	0.0026	0.0041	0.0035
200	0.0050	0.0021	0.0029	0.0025
250	0.0040	0.0020	0.0020	0.0017
300	0.0033	0.0018	0.0017	0.0013

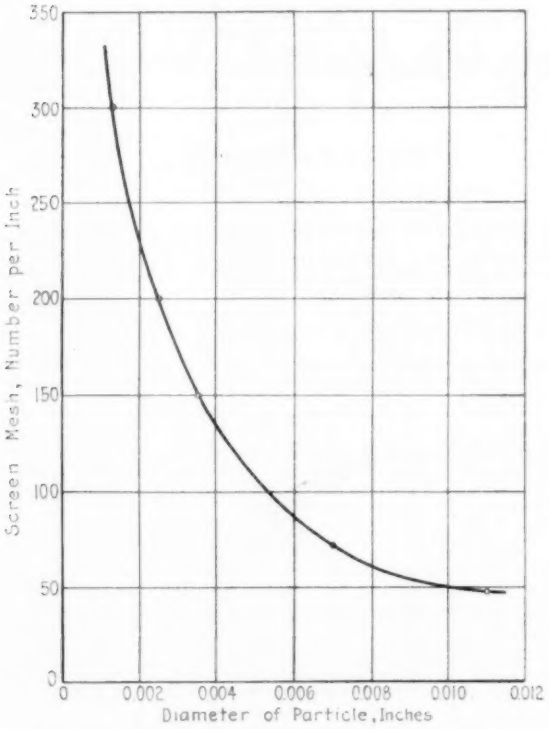

FIG. 23 RELATION BETWEEN SCREEN MESH AND DIAMETER OF PARTICLE WHICH PASSES THROUGH WITH 10 PER CENT CLEARANCE

TABLE 3 MEAN INITIAL DIAMETERS, d_0

Designation Through 200 Mesh, %	Coarsest 33⅓ % Mesh	Coarsest 33⅓ % Diam. In.	Intermediate 33⅓ % Mesh	Intermediate 33⅓ % Diam. In.	Finest 33⅓ % Mesh	Finest 33⅓ % Diam., In.	Mean Diam. d_0, In.
75	167	0.0032	260	0.0016	302	0.0013	0.002
80	185	0.0027	278	0.0014	320	0.0012	0.0018
85	205	0.0024	296	0.0013	340	0.0011	0.0016

APPENDIX NO. 7

TABULATED CALCULATIONS OF μ_1, μ_2, μ_3, AND μ_4

TABLE 4 CALCULATION OF μ_1

Energy Liberation, B.t.u. per cu. ft. per hr.	Surface Arrangement	I_S	$0.93(1-I_S)$	S'_{UC}	S''_{UC}	$0.07\,S''_{UC}$	μ_1
20,000	*A*	0.16	0.78	708	2166	152	0.36
	B	0.18	0.77	874	2166	152	0.38
	C	0.18	0.77	876	2166	152	0.38
	D	0.19	0.76	930	2166	152	0.40
	E	0.20	0.75	966	2166	152	0.41
15,000	*A*	0.17	0.77	832	2166	152	0.37
	B	0.18	0.76	910	2166	152	0.39
	C	0.19	0.75	950	2166	152	0.40
	D	0.20	0.75	976	2166	152	0.41
	E	0.21	0.73	994	2166	152	0.41
10,000	*A*	0.18	0.76	900	2166	152	0.39
	B	0.20	0.74	980	2166	152	0.41
	C	0.20	0.74	1014	2166	152	0.42
	D	0.21	0.73	1060	2166	152	0.43
	E	0.22	0.73	1098	2166	152	0.44

TABLE 5 CALCULATION OF μ_2

Energy Liberation, B.t u. per cu. ft. per hr.	Surface Arrangement	S_{UC} sq. ft.	$a_0 S_{UC}$	$\frac{K_1K_2}{(1-\beta_{RN})K_1+K_2}+K_2$	μ_2
20,000	*A*	400	648×10^{-9}	545×10^{-9}	0.84
	B	800	1295×10^{-9}	883×10^{-9}	0.68
	C	1200	1944×10^{-9}	954×10^{-9}	0.49
	D	1600	2590×10^{-9}	1233×10^{-9}	0.48
	E	2400	3890×10^{-9}	1431×10^{-9}	0.38
15,000	*A*	400	648×10^{-9}	553×10^{-9}	0.85
	B	800	1295×10^{-9}	896×10^{-9}	0.69
	C	1200	1944×10^{-9}	994×10^{-9}	0.51
	D	1600	2590×10^{-9}	1263×10^{-9}	0.49
	E	2400	3890×10^{-9}	1450×10^{-9}	0.37
10,000	*A*	400	648×10^{-9}	565×10^{-9}	0.87
	B	800	1295×10^{-9}	927×10^{-9}	0.72
	C	1200	1944×10^{-9}	1026×10^{-9}	0.53
	D	1600	2590×10^{-9}	1312×10^{-9}	0.50
	E	2400	3890×10^{-9}	1554×10^{-9}	0.40

TABLE 6 CALCULATION OF μ_3

Energy Liberation, B.t u. per cu. ft. per hr.	Surface Arrangement	sq. ft. Surface Exposed	$\frac{dQ_A}{dt}$	Q_A B.t.u. per sq. ft. per hr.	$a_0(T_U'^4-T_C'^4)$	μ_3
20,000	*A*	400	41,000,000	102,500	133,300	0.77
	B	800	50,300,000	62,950	96,100	0.66
	C	1200	53,400,000	44,500	89,700	0.50
	D	1600	58,600,000	36,625	76,300	0.48
	E	2400	61,560,000	25,650	64,200	0.40
15,000	*A*	400	34,500,000	86,333	110,800	0.78
	B	800	42,000,000	52,500	78,900	0.67
	C	1200	43,800,000	36,500	71,000	0.51
	D	1600	47,000,000	29,375	59,600	0.49
	E	2400	50,000,000	20,833	51,300	0.41
10,000	*A*	400	27,240,000	68,100	84,000	0.81
	B	800	31,350,000	39,200	56,200	0.70
	C	1200	33,060,000	27,550	51,150	0.54
	D	1600	35,000,000	21,875	42,300	0.52
	E	2400	37,150,000	15,500	36,150	0.43

TABLE 7 CALCULATION OF μ_4

Total Energy Liberated, B.t.u. per hr.	Surface Arrangement	dQ_F/dt	X_R	dQ_A/dt	μ_4
137,200,000 (20)	*A*	43,500,000	2,500,000	41,000,000	0.30
	B	52,460,000	2,100,000	50,360,000	0.37
	C	55,400,000	2,000,000	53,400,000	0.39
	D	60,350,000	1,750,000	58,600,000	0.43
	E	61,560,000	0	61,560,000	0.45
102,900,000 (15)	*A*	36,700,000	2,200,000	34,500,000	0.34
	B	44,000,000	2,000,000	42,000,000	0.41
	C	45,600,000	1,800,000	43,800,000	0.43
	D	48,500,000	1,500,000	47,000,000	0.46
	E	50,000,000	0	50,000,000	0.49
68,600,000 (10)	*A*	28,440,000	1,200,000	27,240,000	0.40
	B	32,350,000	1,000,000	31,350,000	0.46
	C	33,850,000	800,000	33,050,000	0.48
	D	35,700,000	700,000	35,000,000	0.51
	E	37,150,000	0	37,150,000	0.54

DISCUSSION[1]

GEO. A. ORROK.[2] I was greatly pleased when I received this paper by Professor Wohlenberg, because he gives us for the first time that I have ever seen it in print a coefficient which tells us what proportion of the heat in a pound of coal, or rather the heat evolved in the burning of a pound of coal in a furnace, is absorbed by the radiant surfaces. This is the coefficient μ_4 and I am greatly interested in it. I have plotted in Fig. 24 three rather short lines representing the 15 values of μ_4 which Professor Wohlenberg gives in his paper.

I have been using for some time an empirical formula given by Hudson in 1890 and published in *The Engineer* (London) as being the only available formula which gave workable results. Nicolson and Callendar have given us beautiful radiation formulas, which cover pages of the transactions of some of the English institutes, but when one attempts to work them out he finds that he has to spend five hours calculating one special case, so, therefore, they are of no use to an engineer who is actually doing work.

This formula of Hudson's is short and convenient and I have plotted it alongside of Professor Wohlenberg's μ_4. The Hudson formula is not rational at all, it is purely empirical; but it plots in a smooth curve and the relation with Wohlenberg's rational curves of μ_4 is shown. I call your attention to the difference between rationalism and empiricism.

[1] See also discussion by J. Gould Coutant, page 120.

[2] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.

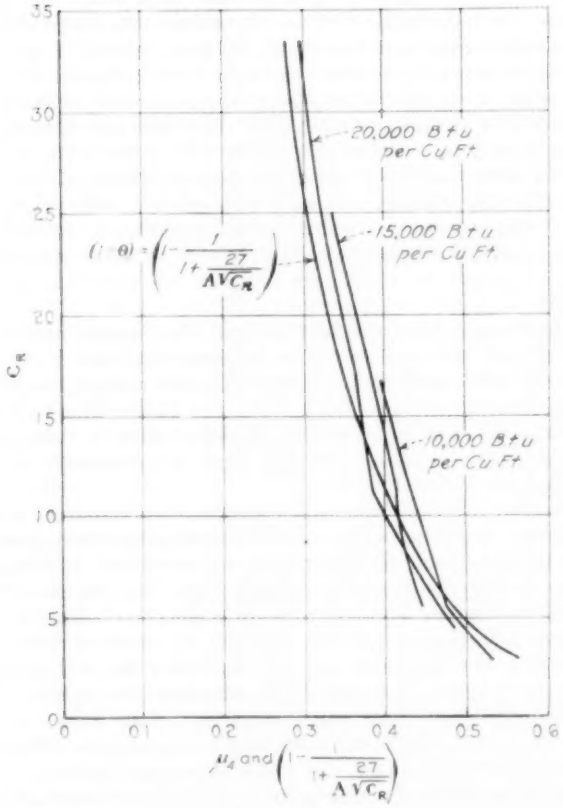

FIG. 24 COMPARISON OF WOHLENBERG'S COEFFICIENT μ_4 WITH THE HUDSON FORMULA MODIFIED BY ORROK

μ_4 = per cent of B.t.u. liberated absorbed by radiant surface.

$\left(1-\frac{1}{1+\frac{27}{A\sqrt{C_R}}}\right)$ = fraction of B.t.u. liberated absorbed by radiant surface.

B. N. BROIDO.[1] The writer has endeavored to figure out the heat absorbed by radiation in a furnace by using the authors' calculations and to compare it with actual results obtained in this furnace. The writer's figures showed μ_4 to be 0.31. The boiler rating was 1800 hp.; the radiation surface in the furnace was 325 sq. ft. The heat absorbed was approximately 60,000 B.t.u. per sq. ft. or 18 per cent. Assuming that the radiation in the lower row of boiler tubes is the same as in the water screen, the total heat absorbed by radiation is 38 per cent or μ_4 is 0.38. The difference between the calculated and actual results is possibly due to the rather complicated calculations and, therefore, to possible errors. It is to be hoped that after more tests with water-cooled walls have been made, it will be possible to correct these factors. It probably also will be advisable to eliminate some factors of minor importance in order to make the calculations simpler and adaptable for general use.

In their calculations the authors give considerable importance to the size of the coal particles. It seems that this is of comparatively little significance and that radiation depends mainly on the radiating energy from the gases of the flame. This is evident from the fact that in a stoker-fired boiler there is considerable radiation from the flame, although there are practically no coal particles in it.

While comparatively little is known about the radiation of solid bodies, this is still more the case with radiation from gases, to which, until recently, no importance whatever was attached. It is now, however, practically certain that the radiation from gases has considerable importance in connection with the heat absorbed in furnaces. The first research on radiation from gases was made about ten years ago. At that time the tests extended only to the exhaust gases from internal-combustion engines. The fact that the heat-transmission coefficient from gases to solid bodies increases considerably with higher temperature differences is due to the radiation from the high-temperature gases.

In studying thermodynamic history, it will be noticed that little importance has been attached to gas radiation. It would seem that we were familiar with the comparatively simple formula for convection and were satisfied with its results, even if in most cases the calculations would not agree with actual results obtained. As a matter of fact, even now while some are endeavoring to develop actual formulas for radiation, Mr. Orrok is proposing a simple empirical formula which he suggests for use in every case. Let us get down the facts and consider each case individually in order to get correct results.

The writer frequently has found that the actual heat absorbed by radiation in a furnace is larger than would be expected when

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figuring the radiation from the fuel bed only. This is no doubt due to the appreciable heat absorbed by radiation from the gases. New data are available showing which gases are radiating heat and to what extent. It may be of interest to find out how these data were obtained. In accordance with Kirchhoff's rule, the radiation of a body at a given temperature is proportional to its ability to absorb heat. Heat as well as light is ether motion, the difference between the two being only in the wave length.

By means of a spectrum the light rays and also the heat rays absorbed by certain bodies can be determined. In this way, therefore, can be determined the heat absorbed by gases, which, in accordance with Kirchhoff's rule, will show also the amount of heat the gas is radiating. This method showed, as mentioned by the authors, that the only gases radiating heat are CO_2 and steam vapor.

Another method exists of determining the radiation ability of the furnace gases, based on the principle of the radiation pyrometer. By reversing the operation, the radiation of the pyrometer can be determined if the furnace temperature is known. The writer understands that this method is being applied to determine the radiation of a furnace, although he never has had the opportunity to use it himself.

Knowledge of the radiation of the flame or of hot gases will not fail to have its influence on the design of furnaces and boilers. Present opinion has stated that in accordance with the requirements for heat transmission by convection, it is necessary for the gases to pass over the heating surfaces at the highest possible velocity, and in streams of small area. Considering, however, gas radiation, the opposite is correct. Where large amounts of gases at high temperatures prevail, it is better to have the gases flow with small velocity and in streams of large area, in order to increase the radiation effect, and due to its longer stay in the furnace, make it possible to give up a considerable amount of heat. The present often-adopted design of boiler with very low gas velocities while the gas is at a high temperature, with increasing velocity while the temperature is dropping, is entirely correct and will probably be used in the future to a greater extent. During the last few years furnace volumes have been increased, which has increased also the capacity and efficiency of the boiler. Present tendencies lean toward decreasing the furnace volume and increasing the velocity of the gases in the furnace where the furnace walls are water-cooled. This would, no doubt, decrease the heat absorption by radiation and also the overall efficiency of the boiler.

Many are of the opinion that large furnace volumes are used only in order to have complete combustion. As a matter of fact, large furnace volumes are also required in order to absorb the maximum heat possible by radiation, with its favorable influence on the overall efficiency.

H. W. BROOKS.[1] This paper is such a real contribution to combustion-engineering literature that one hesitates disturbing in details the excellence of the picture as a whole. By coincidence, the papers presented during this session touch on a common point, namely, the influence of particle size on rate of heat release.

The authors of this paper state, "With present systems, ..., finer pulverization *will probably not increase the rate* of energy liberation." Mr. Andrews[2] has said, "Finer coal will ignite more readily and burn more completely than coarser coal (other factors equal)." Professor Christie in his paper[3] stated, "It is hardly necessary to point out that finely ground coal permits more thorough mixing and more rapid combustion." The latter two statements are apparently at variance with the first one. These latter opinions are supported by the testimony of Etienne Audibert, engineer of the French Bureau of Mines and Director of their Experiment Station at Montluçon, who made specific tests[4] to determine this particular relation with the following conclusion:

> The duration of combustion diminishes with increased fineness of pulverization. This diminution is variable depending upon the character of the combustible. One may in general characterize the order of this by saying that, other things being equal, the combustion of a powder passing through 120 mesh and over 140 mesh, will require about 2 to 3 times the time necessary for a powder of the same nature passing through 220 mesh and over 240 mesh.

Curves of the relative combustion rates with varying finenesses as found by test are given in the reference cited.

To carry this reasoning one step further, were this not the case, there would be little advantage gained in going to the expense of crushing and pulverizing coal, for if combustion were not expedited by it, it would be cheaper to feed the crushed run of mine as received.

One is inclined to doubt, therefore, the authors' statement that "as the particle surface is increased, the oxygen intensity will probably decrease" (presumably in corresponding proportion). Even were this the case, the variation of relative radiance (μ_1) with particle size indicated in Fig. 7 would serve at least in part to account for quicker ignition and an accelerated rate of flame propagation.

THE AUTHORS. In writing such a paper as this the authors had in mind a presentation which would in the main add a little to

[1] Consulting Engineer, Fuller-Lehigh Company, Fullerton, Pa. Mem. A.S.M.E.

[2] Andrews, L. V., A Microscopic Study of Pulverized Coal, *Mechanical Engineering*, vol. 47, no. 5, May, 1925, pp. 429-432.

[3] Boiler Furnaces for Pulverized Coal, see p. 87.

[4] Audibert, E., Etude Expérimentale de la Combustion du Charbon Pulverisé, from *Revue de l'Industrie Minérale*, Jan. 1, 1924.

our understanding of the subject. The criticisms brought out in the discussion are all excellent and justified from certain points of view. Thus Mr. Orrok is in search of an easier method of attack and adapts the Hudson formula, which, however, will undoubtedly require a more thorough investigation before it may be taken as generally applicable.

Both Messrs. Broido and Brooks comment on the authors' statements concerning the influence of finer pulverizatoin on furnace conditions. Mr. Broido apparently is of the opinion that the solid suspended matter has little influence on the radiating powers of the flame. In view of such data as are available this appears to the authors as improbable, as experiments show that the radiating powers of a non-luminous flame, i.e., without solid matter in suspension, are considerably less than those of solid carbon per unit of surface. Perhaps the most recent reliable investigation on this subject is reported in a paper by Haslam, Lovell, and Hunemann.[1] These experimenters found, for non-luminous flames, that there is apparently no direct relation between the radiating power of the mass and its temperature, but rather the radiating power is in some way bound up with the chemical reaction itself. No law is stated to define the exact quantitative measure of such radiations, but it is shown that the Stefan-Boltzmann law does not hold. The data reported also indicate a relatively low radiating power of the gaseous volume as compared to a black body or carbon mass of equal size and at the same temperature. As mentioned in the paper, Callendar's work also shows this to be the case. It appears, therefore, that the amount and fineness of the solid matter in suspension must have considerable influence on the radiating powers of the flame mass as a whole.

Mr. Broido's discussion of the influence of furnace volume on total radiation and how such conditions should be taken account of in the design of steam generators appears to the authors as particularly good.

Mr. Brooks is correct in taking exception to the bare statement, "finer pulverization will probably not increase the rate of energy liberation." If, however, the reader will again carefully study Pars. 66 to 68 inclusive he will find the inference that it is only beyond a certain fineness and with certain methods (those now largely used) of burning such fuel, that a further increase in fineness of pulverization might cease to have an influence in reducing the required furnace volume. It is quite obvious that as long as the relative velocity between the surrounding gaseous mass and the particle can be maintained uniformly high, finer pulverization will increase the possible rate of energy liberation

[1] Radiation from Non-Luminous Flames. *Journal of Industrial and Engineering Chemistry*, March, 1925.

For the larger particles this is true, of course, even when burned by present methods.

Mr. Coutant includes a very valuable way of looking at the furnace problem, namely, from the point of view of the permissible furnace temperature when considering the fusibility of the suspended ash particles. He then goes on to show how much energy as heat must be absorbed from the flame mass before it strikes the first row of tubes, but does not show in detail how this energy is absorbed from the gases. Rather he bases his results on extensive data from practice, which method is no doubt safe and to be recommended, but does not necessarily add greatly to our understanding of the problem. This is of course a main purpose of the paper, and it is hoped that as a result, available data from practice may be better analyzed.

Mr. Coutant's statement concerning the acceptability of the specific-heat determinations of Mallard and Le Chatelier is perhaps true to a limited extent, but there are of course now more accurate data available, some of which are included with the paper.

No. 1957

FACTORS CONCERNING THE ECONOMICS OF SHOPPING STEAM LOCOMOTIVES

By L. K. Sillcox,[1] Chicago, Ill.
Member of the Society

This paper stresses the importance of accurate cost accounting in regard to locomotive maintenance as the basis of the policy to be formulated by the executive in charge of motive power. The author shows the folly of continuing obsolete designs in service, giving concrete cases in which large annual savings were effected by scrapping of inefficient engines. A classification of repairs is given and a tabulation and discussion of the trend of cost of repairs over a long period of years leads to development of the cost of repairs per 1000 tractive-effort ton-miles as a unit by which to measure costs. Other factors which influence policy are the relative frequency of roundhouse and back-shop repairs, the percentage of locomotives in service as related to the number owned, the topography of the country through which the road runs, and, not the least important, the personnel of the repair shop.

THE universal adoption of more or less detailed cost accounts in both large and small establishments and the greater scrutiny given to them by owners and managers are real evidence of the value of such knowledge and the necessity therefor. Such information can be as effectively established and employed in the matter of railroad-locomotive maintenance and be the basis of executive policy with a view toward assisting the average motive-power man in carrying out evident possibilities for economy and an effective procedure at least cost.

2 Economies are of two kinds: internal economies which can be carried out independently by the mechanical officer of his own motion; and the much greater economies which management, as a whole, can only make possible, based on current studies as to policies and possibilities. Though we have emphasized the point of management policy and imagined an executive fixing a complete

[1] Mechanical Department, C. M. & St. P. Ry. Co.

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system, yet no one man, and indeed no combination of men, could evolve a best method off-hand from the foundation. It can only grow gradually, developing here and changing there, in line with economy, as the territory requires or the nature of the traffic and the competition in attracting it become an issue.

3 The serious question we have before us is the necessity of deciding upon an accurate unit of measure. We need correct and simple facts, susceptible of a clear and concise understanding on the part of all concerned, and clear as well as quick application by railway officers charged with the duties of maintenance and operation. Any plan used in the shopping of power involves complete consideration as to necessary maintenance and the effect from characteristics of service, as well as details of motive-power construction, age and availability of shop, roundhouse facilities, etc.

4 Maintenance may be defined as the upkeep or replacement of parts as due, based on individual consideration of each item with respect to limits of wear, strength requirements, and reliability factors. The accounting for the cost of maintenance varies with the type of property. Maintenance of locomotives, for instance, when finally accumulated for the year and reported as required by the Interstate Commerce Commission's classification of accounts, embraces not only the labor and material applied together with the allocation of direct overheads, such as shop expense, power-plant distribution, store expense, etc., but the depreciation and retirement charges. The carrier has the option of determining the rate at which depreciation shall be charged and accumulated currently, but the general principle is to apply a rate somewhat consistent with the estimated total life after deducting the estimated amount of salvage recoverable when the unit is dismantled. Any depreciation which is not sufficient when a locomotive is retired is made up by retirement charges applied against the locomotive-maintenance account at the time of dismantling; therefore equipment maintenance is charged currently with paying off the investment as well as overcoming wear and deterioration. The maintenance of fixed structures, such as bridges for instance, is charged currently as actually incurred, no depreciation being accumulated during the life of the bridge but the entire investment being charged off to operating expenses at the time of final retirement.

5 Taking these facts into consideration, therefore, a fixed structure, having a longer life than equipment, requires a lesser rate of renewals. However, in considering the subject as a whole from an equipment standpoint, we are able to segregate the direct repair cost from other charges, the repairs being confined to a sub-account in the classification, which involves labor plus shop expense, and material plus store expense. It is a fact that in not a few instances much of the expenditure for locomotive main-

tenance, as reflected in the accounts, is occasioned from patching up old construction, even though in some cases this does extend to the point where the actual repair cost, as renewed, exceeds the major portion of the value of the unit as renewed and, consequently, is recapitalized; in which event, nevertheless, the retirement expense must be carried in maintenance.

The Folly of Continuing Obsolete Designs in Service

6 A close study will very often indicate the folly of spending large sums per unit to continue obsolete designs in service for long periods, when the same amount would be more than sufficient, as an initial cash payment, for an up-to-date efficient design of greater capacity and less cost to operate or maintain. As a concrete case, assume that a passenger engine fifteen years old and originally designed to handle a nine-car train is now required to carry fifteen cars on the same schedule. This results in a heavy maintenance cost due to frame breakages, racking of machinery, valve motion, running gear, etc. If the original unit was of 40,000 lb. tractive effort and was costing an average of $9000 per year to maintain, with a relatively low record of 3000 miles per month, it would seem proper to consider a new type of power, say, with 50,000 lb. tractive effort which would afford 5000 miles of service per month and yet not cost more than $5000 per year for maintenance. The saving would be approximately $4000 per year in maintenance cost and the increase in performance 66 per cent. This would justify making a change in power, even though the original unit might have cost $30,000 and the new unit would cost $60,000. The original unit involved a maintenance cost of $9000 per year plus a depreciation charge of $750, or a total of $9750 as compared with an estimated maintenance cost of $5000 per year for the new unit plus $1500 depreciation charges, or a total of $6500. The saving by the new unit would be $3250 per year, to say nothing of additional savings in fuel and transportation expenses. At 6 per cent this recovery would represent an investment of $54,000, or almost the cost of the new unit. If this were done on a large scale the amount of work would be performed with 66 per cent of the number of new units as compared with old ones, and thus the change would be justified. It should be understood, however, that passenger service is more constant than freight, so that this example is not presented with the idea that such a reduction in the number of units is possible in passenger service, but is more applicable to freight service.

7 The extent to which carriers are developed along this line reflects in a general way upon the maintenance policy and the cost. Obsolescence of locomotives is a very indefinite measure and its application, in general, is entirely according to local conditions. A locomotive will grow obsolete and costly to operate even

if it only remains standing, and this same factor will become of increased importance if it is kept constantly in operation for a long term of years without substantial improvement to care for advanced practice, due to transportation losses from excess fuel consumed or questionable reliability in service affecting train movements. For this reason it would seem a wise plan to obtain

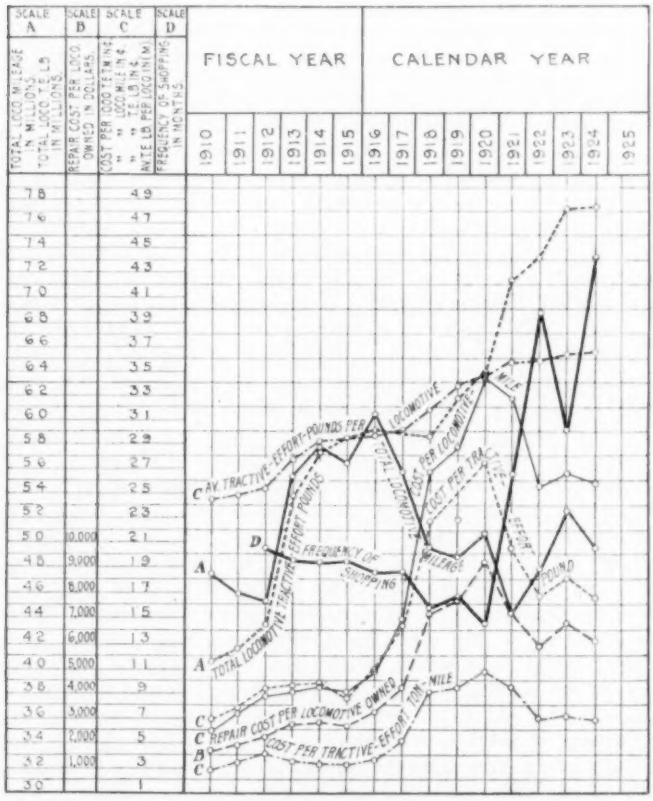

FIG. 1 DATA OF STEAM-LOCOMOTIVE REPAIRS AND PERFORMANCE, C. M. & ST. P. RY., 1910-1924

all the mileage possible from locomotives, so that the time when they are worn out may arrive at least as soon as the stage when they become out of date.

EXTREMES OF POLICY IN SHOPPING LOCOMOTIVES

8 The same idea of promoting efficiency may be applied to the method employed in shopping locomotives, all of which resolves

itself largely into a consideration of the program or the plan employed as best suited by any given administration. The range of possibilities in any given case may be easily determined from a study of practices obtaining on various systems, but, as a matter of fact, there are two extremes, with many variations between them. One is what may be termed the "high-frequency-shopping" and the other the "low-frequency-shopping" policy. The high-frequency-shopping policy is that based on running locomotives through shops with an anticipated service of from twelve to fourteen months with a minimum of roundhouse attention. The low-frequency-shopping policy is that based on running locomotives through shops with the idea of having a service of 24 months or more and with a greater degree of roundhouse attention to attain this length of service. Vital elements in determining such a policy are the relation of the number and size of locomotives owned to the business handled, the road conditions for hauling heavy- or light-tonnage trains, the topography of the country traversed, the distribution of industrial centers, the presence of large terminals, the spacing and capacity of roundhouses, the distribution and assignment of power, the placement of forces as between roundhouses and back shops, the rapidity with which mileage is run out, and particularly the roundhouse and back-shop facilities for handling certain classes of work. Furthermore, where a railroad has back shops of an obsolete character it is practically as well off doing its work in roundhouses, and it may be found helpful under such conditions to construct small modern back-shop facilities at critical points to care for division requirements without increase in overhead expense.

Trend of Unit Costs Since 1910

9 The results obtaining under the two extremes of policy have been observed for a considerable period, and it appears that policy is largely governed by local conditions rather than the reverse. The road with which the author is connected has used both plans. Formerly a high frequency of back-shop repairs was employed, but in 1921 the plan was changed to a low frequency of repairs. The trend of unit costs, etc., both before and after 1921, is illustrated in Fig. 1. The lines plotted represent three general groups, one indicating the growth and size of units maintained, another the various unit costs of maintenance, and a third the frequency of back-shop repairs. The growth of property maintained is represented by the dotted line *A*, which indicates the total tractive-effort pounds owned from 1910 by years to the end of 1924. This growth was not all in the nature of new equipment but represents power added by the acquisition of subsidiary and leased lines as well as some new equipment, and to that extent the growth line

should not be confused with the rate at which new equipment might have eliminated obsolescence. This very fact has a marked bearing on the cost of maintenance, as the total growth consisted of approximately as many of the smaller and older locomotives as of the acquisition of the larger and newer types. The growth of property represents a serious problem in the matter of having back-shop development keep pace with it and of getting continuous use from power. Fig. 1 also shows the increase in the average size of locomotive expressed by the mean tractive-effort pounds per locomotive owned. The size of locomotives is an element in the unit cost of maintenance, and it will be noted that this figure increased from approximately 24,000 tractive-effort pounds per unit to 36,000 in 1924, or approximately 50 per cent.

10 The feature in Fig. 1 which is deserving of closest study is the line *D* representing the frequency of back-shop repairs. This is arrived at by dividing the total yearly classified-repair output into the total owned throughout the year, which expresses the number of years between shoppings thus developed, and this of course varies from year to year according to the difference in the number of locomotives owned or used and the output. This is based upon a classification of repairs instituted during federal control and is translated back to 1912. This classification runs as follows:

CLASS 1: New boiler or new back end. Flues new or reset. Tires turned or new. General repairs to machinery and tender.

CLASS 2: New firebox, or one or more shell courses, or roof sheet. Flues new or reset. Tires turned or new. General repairs to machinery and tender.

CLASS 3: Flues all new or reset. (Superheater flues may be excepted.) Necessary repairs to firebox and boiler. Tires turned or new. General repairs to machinery and tender.

CLASS 4: Flues part or full set. Light repairs to boiler or firebox. Tires turned or new. Necessary repairs to machinery and tender.

CLASS 5: Tires turned or new. Necessary repairs to boiler, machinery, and tender, including one or more pairs of driving-wheel bearings refitted.

General repairs to machinery will include driving wheels removed, tires turned or changed, journals turned, if necessary, and all driving boxes and rods overhauled and bearings refitted, and other repairs necessary for a full term of service.

Running repairs unclassified.

Suffix "A" to any class of repairs will indicate that the repairs are required on account of accident.

"B" will show the initial application of stoker.

"C" will indicate the initial application of superheater.

"D" will indicate the initial application of outside valve gear.

"E" will indicate locomotive was converted from compound to simple or from one type to another.

Mallet locomotives will be indicated by a star following application.

Locomotives receiving Class 1, 2, or 3 repairs must be put in condition to perform a full term of service in the district and class of service in which they are to be used.
Class 4 repairs not less than ½ term.
Class 5 repairs not less than ¼ term.

11 It may be generally conceded that the above is not sufficiently specific to be a complete measure of output since the divisions are not based on work units to a great enough degree to permit of judging shop output in detail.

Cost of Repairs per 1000 Tractive-Effort Ton-Miles

12 The variation in the trend of line *D* is entirely dependent upon the allotment of labor and materials available for maintaining equipment. The shopping frequency increased gradually from 1912 to 1920, and during the latter year locomotives were going through at the rate of once every 14.28 months. In 1921 there was a radical change in the frequency of back-shop classes of repairs subsequent thereto. The frequency of shopping trend, expressed both in years and in months between shoppings, was as follows:

Year	Years between shoppings	Months between shoppings
1920	1.19	14.28
1921	2.20	26.40
1922	3.30	39.60
1923	2.50	30.00
1924	3.70	44.40

The change in plan necessarily brought about some modification in the distribution of machine tools and facilities in back shops and roundhouses. Great care had to be employed to avoid deferred maintenance under such a transition because of the high cost incident to overcoming deferred maintenance promptly and adequately were this condition to have obtained. The roundhouses were partially equipped to do the necessary machinery and running repair work and in some cases rather heavy boiler work so as to properly maintain the power for longer periods. The back-shop forces were reduced in proportion. Prior to the change all judgment as to months good for, miles to be run between shoppings, etc., was based on the theory that locomotives were good for a term of 12 months only. Prior to 1921 there was no specific application of the plan of assigned mileage to be used as a basis for shopping power. This method was put into use at that time and a statement prepared showing the expected mileage to be run out after each classified repair, divided according to types of power. It is important that the same mileage should not be applied to the same type of power regardless of where or how used. In this respect an assigned mileage for each class of service, type of power, and for each division instead of for the system as a whole is necessary in practice, otherwise classified repairs will

be made in roundhouses, but not so reported, in order to avoid breaking the mileage.

13 As to the results obtained from this change of plan, it should be understood that there have been some wage and material price variations since 1920, but these adjustments account for approximately 14 per cent of the reductions attained. The cost trends on the chart merely indicate the actual reductions, with no separation between fluctuations in the cost of material and labor, shop efficiency, etc. The cost per locomotive-mile during the high-wage period of 1920 reached 34 cents when the shopping frequency was 14.28 months, but after the frequency of back shopping was reduced the cost steadily declined and in 1924 was less than 26 cents. This represents a reduction of approximately 24 per cent. The cost per tractive-effort pound was reduced from 27 cents in 1920 to 16.5 cents in 1924, or 39 per cent. The cost of repairs per locomotive owned was $9300 in 1920 and $6000 in 1924, or a reduction of 35 per cent. The cost per thousand tractive-effort ton-miles was reduced from 1.075 cents in 1920 to 0.676 cent in 1924 or 37 per cent. In the meantime the average size of locomotives increased 6 per cent. The method of measuring the cost of repairs per tractive-effort ton-mile was to reduce the tractive-effort pounds owned to tons, multiply by the miles run in thousands, and divide this into the cost of repairs, Account 308. (This unit should not be confused with ton-miles hauled; it is entirely independent of the rate of grade or curvature and is merely the force of one ton acting parallel with the rails and at the circumference of the drivers through a distance of one mile.) Expressed as an equation,

$$D = \frac{C}{\frac{A}{2000} \times \frac{B}{1000}}$$

where

A = total tractive-effort pounds owned
B = total locomotive miles run during year
C = cost of repairs, Account 308, or exclusive of depreciation, retirement, etc.
D = cost of repairs per 1000 tractive-effort ton-miles.

As previously stated, there were reductions in wage rates and cost of material in this period, so that the entire reduction is not to be cerdited to the change in plan referred to.

14 Finally, this principle of shopping locomotives is in no sense special or peculiar. The plan was adopted for the C. M. & St. P. system because it appeared to be properly applicable. It is a matter of interest in this connection, however, to make a study of ten carriers where there is a wide range of policy, using the same units outlined above. The value of the units used cannot be considered as entirely intrinsic and for that reason the method

employed should not be considered as absolute. The chart of Fig. 2 is a result of a study of various carriers, some having a high-frequency and some a low-frequency method. In plotting the data, scales were used merely to throw relative items together in order to indicate those which run in a certain ratio and those which run inversely. The horizontal scale is the average tractive-effort

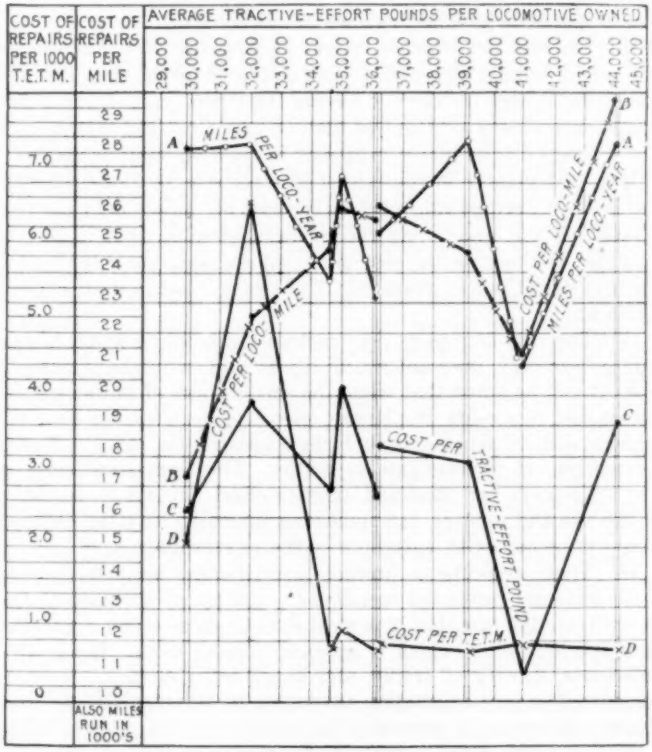

FIG. 2 COMPARISON OF COST OF MAINTAINING STEAM LOCOMOTIVES CONSIDERING AVERAGE SIZE, AVERAGE MILES, AND COST OF REPAIRS PER MILE PER TRACTIVE-EFFORT POUND AND PER 1000 TRACTIVE-EFFORT TON-MILES

pounds per locomotive owned — to show the size maintained. The data are based on 12 months in 1924.

15 Another point may be observed. In the matter of performance the average miles per locomotive run per year is given by line *A*. This shows a difference in performance which does not follow relatively the variation in size of power, indicating many degrees in intensity of use, etc. The mileage ranges from 20,800

to 28,500 per locomotive per year. The carrier with the largest size of power made practically the same mileage per locomotive as the one with the smallest size of power shown on the chart, whereas, those administrations with locomotives of a size ranging between the two extremes made less mileage per locomotive. This may be considered an element of performance and demand characteristics of the lines involved. In this case the highest mileage was 37 per cent greater than the lowest shown.

16 The cost per locomotive-mile as shown by line *B* is low for the carrier having the smallest size of power, being less than 17.5 cents, and is high for the one having the largest size of power, the upper range being a little less than 30 cents per locomotive-mile. The average size of power expressed in tractive-effort pounds was 47 per cent greater in the largest average size as compared with the smallest average size, whereas the cost per mile was 71 per cent greater for the larger than the smaller power, so that it may be said that in this case the cost increased one and one-half times as the unit size of power increased. This figure is not in itself entirely significant.

17 The cost of repairs per tractive-effort pound being a measure of size only, does not run in direct ratio with the increase in size. It will be noted that while the smallest size of power has the lowest cost of repairs, the largest size of power does not have the highest cost of repairs, both being exceeded by carriers having locomotives of intermediate capacities. The composite factor of cost of repairs per thousand tractive-effort ton-miles, consisting of the size and mileage, ran somewhat inversely with the capacity of power owned, with variations in the intermediate sizes.

18 There would be no profit in going further into details, since these data are presented merely as a matter of comparison and to illustrate the fact that it is most difficult to establish a criterion or a specific policy which is applicable in all cases. Road characteristics, density of traffic, and many other elements must be considered in any calculation that aims to be complete and conclusive. However, in general it has been found that carriers using large-size power have a fairly high frequency of back shopping, which is of great interest in this study because it is evidence, for the most part, of intensive use and a more or less uniform power demand.

19 These data indicate that as the average tractive-effort pounds owned increases, the cost of repairs per mile increases in a rather definite way; that the average miles per locomotive run and the average cost of repairs per tractive-effort pound have the same range characteristics; and that the average cost of repairs per thousand tractive-effort ton-miles varies somewhat inversely with the average tractive-effort pounds per unit owned, with intermediate fluctuations.

20 In using tractive-effort ton-miles as a unit of measure in this case it should be understood that there is a variable feature in the actual amount of capacity employed as compared with the maximum available. The larger units may or may not exert as great a percentage of the total tractive effort as the smaller units, and to the extent of this variation this measure must be considered only with the knowledge thereof. Here again maximum or ruling gradients, scheduled tonnage, etc., are factors showing how near the capacity available is utilized as compared with the maximum at hand. This unit of measure does not include the work performed, that is, the tonnage hauled, which if added would doubtless give trends differing from those shown on the chart; but the tonnage hauled is a matter of record in freight performance only, whereas the data thus far considered apply to all classes of power. Statistics are now available to show the gross ton-miles per freight-train-hour, which is a fair measure of locomotive performance when considering the average size as well as the number of units employed. This reflects efficiency in train handling as well as corresponding utilization of power, and where there is prompt turning in roundhouse care and running repairs fewer locomotives may be used for a given traffic, thus building up a reserve and maintaining a proper shopping period. It is necessary, of course, to apply such data to road engines only, and this can be done with the information at hand. The variation among carriers in such performance is quite extreme, some having as high as 26,000 and others as low as 12,000 gross ton-miles per train-hour. This difference is far greater than the average size of power and other influencing factors such as grades, curvature, signal stops, etc., would indicate, and leaves open to study the question of greater development along the line of utilization of power and consequent shopping methods.

Effect of Specific Shopping Policy on Cost of Repairs

21 Just what effect the specific policy followed has upon the trend of the cost of repairs is not thoroughly ascertainable, but it would seem that a minimum cost may be expected when it has been definitely established that a proper balance between classified and running repairs has been arrived at. No specific formula for reaching this division has yet been developed. A study of the cost of repairs per mile divided between running and classified work does not present a solution, because whenever it is necessary to decrease or increase forces and expenses, the fluctuation is felt directly in the back shop and only indirectly in the roundhouse. Running-repair costs follow a more uniform trend, and the policy of the extent to which engines are given running repairs in roundhouses must be fashioned according to the relation of road time to time allowed for turning, which is largely affected by the units

available for service call. Running repairs are not so directly affected by the disposition of revenues as classified repairs except where the carrier is in a position to enjoy revenues which exceed the expenses to such a degree as to constantly insure earning the net requirements.

22 One way of measuring performance with shop output is to compile the locomotive-miles run since last shopping, setting this up in a cumulative way monthly and comparing it with the miles restored in classified repairs. The miles restored must necessarily be based on the assigned mileage, and the balance between miles run out and restored cannot be accurately judged unless the assigned mileage is practically correct and consistent with actual performance. When the assigned mileage is set too high, then there will be more restoration of mileage reported by shop output than is actually run out; on the other hand, if the assigned mileage is too low, then the restoration is not in keeping with the run-out mileage. The experience of the C. M. & St. P. along this line has developed the fact that the application of the assigned mileage for each type of power, regardless of its use, is not sufficiently specific, and such process, when employed, needs revising by developing an assigned mileage for each type of power and for each division so that the local characteristics as to track, curvature, gradients, service, consequent tire and lateral wear, and boiler repairs can be considered as factors. Theoretically it is possible to group repairs into time cycles, but time is only one of the elements of shopping power, and as the frictional wear is mostly overcome in roundhouses the time element for shopping power resolves itself very largely into cycles based on necessity for heavy boiler repairs.

23 The factors used by the author's road in the general plan of shopping are assigned mileage, time, and actual physical condition based on customary and frequent inspections. Where there is a low rate or run-out mileage, time enters into the calculation to a greater degree than where mileage is run out rapidly. Physical condition is a vital element to overcome the differences in divisional characteristics or variations in the service rendered according to track, water, and other features. In addition, it can be arranged to set a limit to expenditures for the various types of power according to the class of repairs to be given, any overexpenditure to be reported and explained so that extraordinary repairs may be a matter of record. The regulation of frequency of repairs to locomotive parts is a matter of long-range study and the cycling of repairs cannot be followed specifically in all cases, although it has been the company's general policy to endeavor to follow a class No. 3 repair with a No. 4 or No. 5 and then with another No. 4 or No. 5 followed by a class No. 3, in other words, having two minor repairs between two major repairs; but this depends upon the nature of service performed and the severity as well as the rapidity with which mileage is run out. It cannot at

the outset be presumed that we can obtain approximately 50 per cent of the mileage between classified repairs for the first class No. 3 repairs and then 25 per cent for each of the minor repairs, as experience has developed that inasmuch as taking up lateral, tire turnings or changes, etc., constitute a classified repair, tire and lateral wear develop and become due for renewal just as soon after a class No. 3 as after a class No. 4 or No. 5 repair; so that it is well to contemplate a distribution of the mileage and make it equal for each class of repairs, whether a No. 3, No. 4, or No. 5 class, but regulate the cost of each shopping accordingly.

A Study of Enginehouse Operation

24 A study of shop and roundhouse facilities cannot be summarized in terms of specific units so as to outline the direct effect upon policy and the results thereof. It involves the number of roundhouses per mile of track, the frequency of turning locomotives or the miles run between turnings, etc. As a matter of interest, it may be stated that during the time in which the change in policy referred to was taking place, studies were made of enginehouse operation not only as to repairs made but as to the cost of turning power, frequency of turning, etc., with results shown in the table immediately following.

	Transportation expense (not maintenance)		
	1922	1923	1924
Average turnings per month	53,572	59,784	54,911
Average cost per month	$430,492	$384,969	$331,830
Average cost per engine turned	8.30	6.44	6.04
Average cost per engine-mile, cents	10.25	8.35	7.68
Average miles run between turnings	78	77	78

It is apparent that the frequency of turning power decreases with the increase in average distance between roundhouses, and that the average miles per engine turned will increase with the spread in distance between roundhouses.

25 A statement of the actual hours of service per engine per day reflects the intensity of use and efficiency in turning, all of which depends largely upon the condition of power, the amount owned, and the character of maintenance. We have not been able to observe any adverse effects on the turning of engines because of decrease in frequency of back-shop repairs; it might have been assumed that so low a frequency as was developed in the change of policy would throw too much of a burden upon roundhouses for intermediate repairs and increase the hours of detention. The cost per mile of running repairs incurred in roundhouses is greater than the cost per mile of classified repairs made in the back shops, the former running rather uniformly with the number of engines turned and the latter fluctuating more in relation to increase or decrease in operating expenses because of regulating them according to revenues.

26 The formulation of a specific policy of back-shop attention to power, therefore, must consider a wide scope of performance, embracing the amount of property owned in relation to the demand and use thereof, the type and size of power, shop and roundhouse facilities and their relation to each other, road conditions, rapidity with which mileage is run out and restored, the ability to maintain a minimum but uniform force in back shops throughout the entire year, the ability in addition to increase the rate of repairs in the low-peak-loading months so as to have a larger supply available for service in high-peak months, the range of variation between high- and low-peak months, etc. Where back-shop and roundhouse facilities are modernized and properly regulated by means of machine tools, power plants, handling devices such as cranes, hoists, tractors, manufacturing devices, and an adequate store stock, it should be possible to determine closely the final balance between classified and running repairs with its relation to the cost of performance expressed in various units of measure, some of which have been outlined above.

Bearing of Utilization of Power on Shopping Policy

27 The utilization of power requires careful and detailed study because any variation must of necessity have a marked bearing upon shopping policy. It has been found when comparing the performance of a greater number of carriers than used in plotting Fig. 2 that the tendency to run out locomotive-miles rapidly increases somewhat with the increase in size of power, and the rate at which mileage is run out is a determining factor in the proper frequency of classified repairs. There is a great variation in the mileage performance, and this reflects either good or poor power assignment and use, or a high or low density of traffic. Where the percentage of locomotives required for business is large in relation to the total ownership, where the monthly mileage is high, and where the hours of service per day are above the average of eight, this reflects a small reserve for detention in and awaiting shops, and the back-shop efficiency is then a more vital factor and the time consumed in making repairs is a greater element than where opposite conditions prevail. The average number of days out of service for the item of in and awaiting shop is not a matter of uniform record and therefore is not always available, but it is important to have for this purpose a detailed knowledge of the number of days locomotives are out of service. It is possible to arrive at the average days of detention in the past so as to regulate the future by the use of the following formula:

Let A = average number of locomotives in and awaiting shop per month
B = total locomotives owned
C = total days in the year and
D = frequency of shopping.

Then

$$\frac{A}{B} \times C = \text{average number of days detention per locomotive per year}$$

and

$$\frac{A}{B} \times C \times D = \text{number of days detention per shopping.}$$

28 In cases where the percentage of power in service is low as compared with the total owned it is usually found that the hours of service per day are low or approximately six, which makes it possible to have a large waiting list and to turn locomotives less rapidly through the shops and still have ample protection for service. It would be expected that where the average size of power was small there would be more units out of service than where the average size of power was large, other things being equal. On the other hand, the data supporting the chart indicate that where large power runs out mileage as rapidly as small power it incurs a greater frequency of back-shop repairs, probably due to terminal facilities not having been improved to the same extent as the motive power. The tendency should be to increase the utilization of power, and increased utilization will depend largely upon its condition and the degree of maintenance. When longer runs are installed and fewer locomotives are used for a given service, this naturally builds up a surplus. With a surplus of power the shopping problem is less acute. Where the gross ton-miles per train-hour indicate a slow movement in a territory where traffic is dense, trains should not be delayed because of short runs made by locomotives and frequent delays due to changing power. Movement in this case requires acceleration. To overcome this trouble it is necessary not only to increase the length of locomotive runs but to eliminate intermediate terminals and classify trains so that the local service will be confined to the fewest possible trains, permitting a greater number to proceed without such interruption. Intensive service requires a revision of the method of repairs, and in some cases calls for a higher standard of repairs than where more locomotives are used in the same service. The objective should be to get more locomotive-miles per month out of fewer active locomotives and more hours of service per day out of each locomotive; then other factors will naturally increase the gross ton-miles per train-hour.

29 Turning to the problem of the time element applied to locomotive output, where orders are issued to shops for locomotives of a certain character to be made ready for service on short notice, the time factor, as regulating the date of delivery, is very often a matter of supreme importance. One of the outstanding factors is that of the machine-tool and material-handling equipment, with which is identified the means of producing work

rapidly and to the best advantage. Another equally vital point is that of the organization of the work, which must, if time is to be saved, be planned on a definite and progressive system. All these measures are applicable to old as well as to the most modern new shops; however, they are likely to lose much of their value unless the workmen themselves coöperate in the avoidance of time wastage. Piece-work and unit-output-measure systems are designed to make wasting of time on the part of the workmen unprofitable to them, but even where these are in force it is often found impossible to correct the loss. It is therefore only through the strict coöperation of every one concerned, combined with proper shop systems and the use of tools and facilities fully adapted to their purpose, that rapidity of production can be assured.

Conclusion

30 The designated mechanical officer should place before his executive a practical and sound analysis of the requirements for the upkeep of motive power, shops, power plants, and tool equipment, and recommendations for expansion or rehabilitation to save against existing unit expense known to be out of keeping with the possibilities for best achievement. It is also desirable to view the problem from the standpoint of (1) Performance, (2) Cost, and (3) Progress.

31 In making submissions for proposed expenditures they may be grouped under items or savings as dealing with:

1 Fuel
2 Labor
3 Materials
4 Delay to Train Movements
5 Accident Prevention or Personal Injury.

Once we have grasped these fundamental facts, we can promptly get rid of not a few popular fallacies already referred to. It is regrettable and not a little surprising that in fixing ordinary running expense and upkeep the budget allowance to be approved by any administration should fail to be based upon the requirements as made necessary because of traffic or operating conditions and complying with a general policy formulated to properly care for conditions in the most effective and economical manner, and not altered to fluctuate with current temporary drops in traffic, which are often seasonal. Where there is an increase or decrease in business handled during a long period it is necessary to modify the assignment of forces employed in such a way as will permit of increasing forces economically, that is, gradually, if materials and facilities are in readiness to absorb the advantage of greater forces to be employed, and to decrease them without permitting the property to depreciate or build up deferred maintenance. Such changes in the budget should never be made without the most

careful consideration of all phases of the problem, based on considerations of a broad policy.

32 For instance, during the course of five or ten years the traffic on a given property may warrant certain expenditures for maintenance of locomotives, shops, power plants, tool equipment, and new locomotives; this expenditure should be made constantly each and every year. The policy of trying to maintain an even or favorable allocated net earning capacity each month, regardless of the business handled, in order to save against financial fluctuations as to credit or stock issues, acts to cause a considerable influence to make administration difficult, particularly during low seasonal movements. An accumulative allocated net trend, with a full knowledge of balances between losses and gains, should permit of a more even flow of maintenance work. Unless this is done a stable policy cannot readily be worked out or maximum economy obtained with a minimum outlay for facilities and for labor.

33 What should be arrived at is a reasonable maintenance standard for the property as a whole. The basis should be gaged from a consideration of the general earning capacity and also upon the kind and class of traffic handled. With this part correctly fixed, slight fluctuations in business should not materially alter the yearly progress as applied each month with the view toward stability of employment and proper discipline and administration of the work. In a large sense the motive-power man is only as successful as his management will let him be. This is one of the reasons why so many able men have left the railroad service for other and more productive fields. Every motive-power man of wide experience knows what a lack of executive support or full coöperation means. The best policy on earth avails little if it does not have support from the top of the organization down. Too many of our executives judge the mechanical man by locomotive-miles produced without any direct reference to work done by the locomotives, which is a vital factor. It is for this reason that data were presented here to indicate not only mileage performance, but a composite factor of mileage and size of capacity.

34 One other point in conclusion: It has frequently proved to be the case that, regardless of all of the devices and methods employed to increase the efficiency of power, the personnel and organization employed are the most important in this respect. There is a great difference of opinion as to whether or not a locomotive is due for shopping. Judgment in this respect varies, but the basis depends largely upon training and the policy adopted, whether expressed or implied. The psychology of high or low frequency varies. The human factor in shopping policy is as vital as the material factor.

DISCUSSION

L. P. MICHAEL.[1] This paper has brought out many points of interest to railroad men in charge of the shopping of locomotives and maintenance. One of these is that perhaps the reason for shopping, and making the time between shoppings of locomotive longer than was formerly the practice, is due to the change in locomotive design. The locomotive of a few years ago was a very simple machine. At the present time it is not a simple machine, in the sense of the word, for the reason that the locomotive itself has in connection with it a number of devices and appliances which must be maintained in serviceable condition. These require a great deal of inspection and work to maintain them.

The locomotive of a few years ago was the standard eight-wheel type. It has changed gradually to the ten-wheel type and then to the modern types that we have at the present time. During this change a great many details have been added, such as stokers, power reverse gears, superheaters, feedwater heaters, and other devices which require systematic periodic inspection. This work does not necessarily mean that the locomotive must come into the shop; it is of such a nature that it can be done in the enginehouse, and this the writer believes is one of the reasons why the extension of time was made greater than formerly.

WILLIAM JOOST.[2] The writer believes that there is no subject more important to railroad officials at this time than the shopping of power. A part of the problem is the best practice of locating the shops for heavy repairs. The best plan probably would be to have a centralized shop for very heavy repairs, that is, for such repairs as require many and expensive tools. An alternative would be to have most of the locomotives repaired locally in small shops. In that way we get better results and avoid a dead line and extra expense in moving the locomotives back and forth to the main shops.

The writer believes that a locomotive frequently can be repaired in the roundhouse more cheaply than in the back shop, especially if the same organization is repairing the locomotives that has to maintain them while they are in service. There are many weak spots that a roundhouse organization will find and repair in a locomotive that probably the back shop organization would overlook. For instance, a certain type of switch engine that we have been repairing in a roundhouse continually has given much trouble due to castings, etc., working loose, and in some cases being lost on the right of way. After the roundhouse forces undertook

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[2] C. M. & St. P. Ry., Milwaukee, Wis.

repairing these locomotives, they were very careful to see that they were properly put up, and they found some means of holding the weak parts.

Another good feature of repairing locomotives in roundhouses that probably would not be a good practice in back shops, is that the roundhouse organization knows just exactly what a particular locomotive requires. The roundhouse maintained it while in service, and will not do any work that is not necessary. Such practice lowers the cost of repairs. This is not an argument against thoroughly going over every part in the back shop, but there are a good many things that a roundhouse force will save. Thus, the roundhouse force that is away from the back shop and cannot get material as easily and has no system of following material, will be very careful in looking over its material and in saving every bit of that material which possibly can be saved. For instance if an engine needs driving boxes, if the roundhouse force could save one out of a pair, they would save it. The same thing would apply to rod bushings or tires, for if the tires were worn 3/32 of an inch the back shop would probably turn them, while the roundhouse force would probably let them go and get another year of service out of those tires.

The author brought out the importance of having a good personnel. This is probably just as important and more important than having good equipment with which to work. That will bring home to us again the importance of having local shops. The local men or the division officers that have charge of the power, if they are called upon to repair their own power, probably will take much more interest in increasing the mileage of the engines and in making those repairs than will a centralized shop. It often happens, with a centralized shop, that local divisions are crowded for room or help and are not able to overhaul their power properly. They then find some means to get their power to a centralized shop; whereas, if they were called upon to make most of their repairs or the ordinary repairs at the local shops, much better results would be obtained.

Another very live subject, especially due to the fact that we are called upon to make more mileage with our locomotives than formerly, is the picking out of the weak points. Probably all railroads have certain types of engines from which they expect to obtain a certain number of miles. It is difficult to do this without overhauling the locomotives in the roundhouse between shopping periods. It thus becomes important to find the weak points and strengthen them to enable the locomotive to make the assigned mileage. This procedure has been carried out successfully on the railroad with which the writer is connected, and some of the locomotives which were able to make only 30,000 to 40,000 miles between heavy repairs are now making 70,000 miles very easily.

J. A. ANDERSON.[1] On the Chicago, Milwaukee & St. Paul Railway, the best results have been obtained by giving the locomotives a thorough overhauling when they are sent to the back shops, irrespective of the so-called classification of repairs. All parts of the locomotives are gone over, checked to blueprint, and brought back to standard to insure full term of service after shopping. Frames and boilers are rechecked for proper alignment, considerable care being paid to cylinders and valve bushings to insure true circle. All driving wheels are rechecked for quartering and counterbalancing, and axles and wheels machined perfectly round; this, together with bringing the valve-motion parts, spring rigging, and brake rigging back to standard, will be reflected in the roundhouse maintenance.

We have found it pays when locomotives are in the back shop to check the counterbalance and quartering on every pair of wheels. When locomotives are turned out with driving wheels, journals, and crankpins round, and frames, wheels, cylinders, valve-motion parts, brake rigging, and spring rigging brought back to standard, roundhouse repairs will be reduced. Too often locomotives do not give the same service after general repairs as they did when new. If practices as outlined are followed, the locomotive will give the same character of service after being rebuilt in the back shop as it did when it was received from the builder.

By setting repair standards high, it is only natural for the repair cost to appear high — in fact, higher than when locomotives are repaired under conditions where the standards are not so exacting. If the high maintenance standards as outlined are followed from year to year, after a period of three years, and sometimes less, the economy will be apparent, the period between shoppings being greater, serviceable hours increased, and running repairs, engine failures, coal consumption, and all locomotive repairs decreased.

There is one other point in the paper to which the writer will refer. The railway companies of this country excel in the production of transportation. In order to produce any item on a production basis, taking into consideration the keen competition existing, it is necessary that considerable care be given in the selection of tools and the continuance of their use. The automobile companies having made the greatest strides in this direction do not hesitate to retire a tool as soon as some other tool can be found that will justify the expense of replacement. There is an opportunity for the same rule to be applied to the locomotive, it being the tool the operating department of the railroad company uses to produce transportation. There are a number of locomotives in service today that are expected to produce transportation on a production basis that should be retired and replaced by new tools that will give the service which may be possible.

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H. T. BENTLEY.[1] Repairing locomotives is a rather peculiar proposition, especially with the varying amount of business that has to be taken care of. Three or four years ago when we were doing a very heavy business, the question came up as to whether it was not desirable to enlarge our shops in order to be able to repair more locomotives. After a careful study of the situation it was felt that the shops we had were sufficient for the normal condition of business and it was decided that the roundhouses (some of the more modern ones) could be given sufficient help and with the facilities that they had could come to our assistance.

We told the men that we were just going to turn over some work to them to find out whether they were qualified and able to turn it out economically and thus help us out in this particular stress. They responded wonderfully and much work was done in roundhouses; not perhaps as economically as we could have done it in the shops under ordinary conditions, but the need for more power at that particular time was so great that we were glad to be able to turn to the roundhouses and give them some of the less important heavy repairs to make.

We should remember that the hauling of locomotives from the one end of the road to the other is very expensive. In our case we can haul them 1200 or 1400 miles, and it costs just as much to haul a locomotive to the shop as it does to haul an equal weight of cars to the same shopping point.

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STRESS CONCENTRATION PRODUCED BY HOLES AND FILLETS

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In many practical cases a very high stress concentration is produced by holes, grooves, notches, and sharp variation of cross-sections. This stress concentration is particularly undesirable where materials undergo reversal of stress. Coefficients of safety as hitherto employed have been large enough to cover such weakness. The modern tendency in design is to increase working stresses, thus obtaining lighter and more economical structures, and concentration of stress is thus receiving more attention.

The authors attack a number of practical problems relating to holes and fillets, and employ several methods of approach; namely, analytical method giving approximate solutions for certain simple cases; photoelastic method by means of which stress concentration can be studied if the problem is a two-dimensional one; Lueders' Line method, the study of certain lines appearing on the polished surfaces of a mild-steel specimen subjected to simple tension or compression, at the points of highest stress concentration; and fine extensometer measurements at points of stress concentration. Analytical and experimental methods are found by the authors to be in satisfactory agreement as far as static loads are concerned. When reversal of stress is involved, the weakening effect of stress concentration can only be established accurately on the basis of fatigue tests.

IT IS A well-known fact that holes, sharp reëntrant corners, and rapid changes of section produce high stress concentration and are undesirable in machine parts, especially when these parts are subjected to reversal of stresses. There are several cases where this stress concentration can be calculated analytically, also cases

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in which the stress concentration can be determined by experiments, but until now very few attempts have been made to bring these data into account in actual design. This is due principally to the fact that the coefficients of safety which are applied in design are usually large enough to give sufficient margin for any stress concentration which may occur in service.

2 With the modern tendency, however, to increase the working stresses and to obtain in this way more economical structures, the problem of stress concentration becomes of great practical importance, and its effect must be considered in actual design. Before this can be done, however, it is necessary to know something about the magnitude of the stresses involved. In the following pages different methods of solving such problems on stress concentration are discussed and some practical conclusions are made on the basis of these solutions.[1]

I — Methods of Estimating Stress Concentration

3 Several methods can be employed for studying stress concentration, and these are discussed in the following paragraphs.

(*a*) *Analytical Method.* This method enables us to calculate the stress concentration for such simple cases as circular or elliptical holes, on the assumption that the material is homogeneous and perfectly elastic. An approximate method of solving problems of this kind will be developed later.

(*b*) *Photoelastic Method.* By using this method the stress concentration produced by holes or fillets can be studied if the problem is a two-dimensional one. Its application to the solution of the fillet problem will be given later.

(*c*) *Lueders' Line Methods.* Certain lines, known as Lueders' Lines, appear on the polished surface of a mild-steel specimen subjected to simple tension or compression when the stress attains a certain value. This property can be used for detecting the points of highest stress concentration and for evaluating the maximum stress. Application of the method to the fillet problem will be considered later.

(*d*) *Fine Extensometer Measurements* can also be used for determining the stress concentration produced by fillets and holes. Certain experiments of this kind have been made by German investigators,[2] and the results obtained are in very good agreement with the analytical solution.

[1] Only the problems of stress concentration in structures under tension, compression, or bending are considered. The problem of stress concentration in shafts under twist will be considered later.

[2] See articles by Professor Preuss in *Zeit. d. Ver. deutscher Ing.*, 1912, p. 1349, and 1913, p. 664.

II — Circular Holes

4 If a plate with a small circular hole is subjected to uniform tension (see Fig. 1), the stress distribution near the hole may be obtained from the general equations of the theory of elasticity. An approximate solution of the same problem can be obtained by using the following method. It is known that a hole causes a redistribution of stress in its immediate neighborhood only, and if a circle of diameter D is considered as in Fig. 1(a), the stresses at points on the circumference are not materially affected by the presence of a hole, provided that D is large in comparison with the diameter of the hole d. Imagine now that the portion of the plate bounded by the cylindrical surfaces of diameters D and d is cut out from the plate as shown in Fig. 1(b) and that the internal forces acting in the plate at points on the circumference of the diameter D are applied to the outer edge of the circular plate. These forces are acting in the direction of tension and their magnitude at any point m is given by the known equation

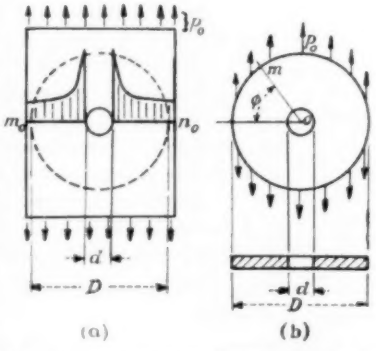


Fig. 1 Analysis of Stress Distribution in a Plate with a Small Circular Hole and Subjected to Uniform Tension

$$p = p_0 \sin \phi \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

in which p_0 is the tensile stress uniformly distributed over the end section of the rectangular plate, Fig. 1(a), and ϕ is the angle between the radius om and the horizontal axis.

5 Now the problem of the stress concentration produced by the hole is reduced to the calculation of stresses in a circular ring [Fig. 1(b)] subjected to the action of given external forces. In discussing this problem we take symmetry into account and consider one quadrant only (Fig. 2).

Let M_0 = bending moment on the horizontal cross-section m_0n_0 of the ring

S_0 = longitudinal force on the same cross-section

M and S = corresponding quantities for any cross-section mn

ϕ = angle between any cross-section mn and the horizontal cross-section m_0n_0

$h = \dfrac{D-d}{2}$ = depth of cross-section of the ring

t = thickness of the ring

$\rho = \dfrac{D+d}{4}$ = radius of the center line of the ring

r = radius of the neutral axis during bending of the ring

γ = distance of the neutral axis from the center of gravity of the cross-section

$A = th$ = area of cross-section of the ring

$S = A\gamma$ = first statical moment of cross-section about neutral axis

z = distance of any point of the cross-section from the neutral axis. This distance is considered as positive in the direction from the neutral axis toward the center of the ring

p_0 = uniform tensile stress in the plate

p_1 = tensile stress produced by the longitudinal force S_0 in the cross-section m_0n_0

p_b = bending stress

p_2 = maximum bending stress in the cross-section m_0n_0.

6 The longitudinal force S_0 can be calculated from the following equation of statics:

$$S_0 = \frac{p_0}{2} Dt \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [2]$$

and the corresponding tensile stress will be

$$p_1 = \frac{S_0}{A} = \frac{p_0 D}{2h} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [3]$$

7 The value of the bending moment M_0 in the cross-section m_0n_0 (Fig. 2) can be calculated from the condition that, as a result of symmetry, the cross-section retains its horizontal direction during deformation. This condition can be written as follows:

$$\int_0^{\frac{\pi}{2}} \frac{M d\phi}{EA\gamma} - \int_0^{\frac{\pi}{2}} \frac{S d\phi}{EA} = 0 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [4]$$

where the first member on the left represents the rotation of the cross-section m_0n_0 produced by the bending moment M, and the second is the rotation due to the longitudinal force S. Considering all the forces shown in Fig. 2, it can be shown that the longitudinal force S and the bending moment M for any cross-section mn have the following values:

$$S = \left[S_0 - \tfrac{1}{2} \int_0^{\phi} Dtp_0 \sin\phi d\phi \right] \cos\phi = \tfrac{1}{2} tp_0 D \cos^2\phi \quad . \quad . \quad [5]$$

$$M = M_0 + \tfrac{1}{2} p_0 tD(1-\cos\phi)\left(\tfrac{1}{4} D(1-\cos\phi) + \frac{h}{2} \cos\phi \right) - \tfrac{1}{2} p_0 tD\rho(1-\cos\phi) \quad . \quad . \quad . \quad [6]$$

8 Substituting [5] and [6] in Equation [4] and performing the integration,

$$M_0 = \frac{p_0 t D^2}{2\pi}\left[\frac{2\rho}{D}\left(\frac{\pi}{2}-1\right)+\frac{\pi}{2}\frac{\gamma}{D}-\frac{h}{D}\left(1-\frac{\pi}{4}\right)-\frac{3}{8}\pi+1\right] \quad [7]$$

9 For a rectangular cross-section,

$$\gamma = \rho - r = \rho - \frac{h}{\log_e \frac{D}{d}} \quad \ldots\ldots\ldots [8]$$

The bending stress at any point in a cross-section of the ring is given by the known equation of the theory of curved bars, namely,

$$p_b = \frac{Mz}{S(r-z)} \quad \ldots\ldots\ldots [9]$$

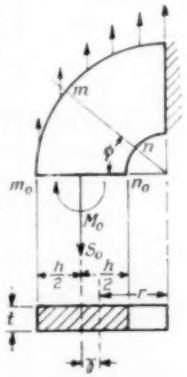


FIG. 2

10 The maximum bending stress in the cross-section $m_0 n_0$ takes place at the point n_0 (see Fig. 2) where

$$z = \frac{h}{2} - \gamma \text{ and } r - z = \frac{d}{2}$$

Substituting this in Equation [9] gives

$$p_2 = \frac{M_0}{\gamma d t}\left(1 - \frac{2\gamma}{h}\right) \quad \ldots [10]$$

Now by using Equations [7] and [8] the maximum bending stress can easily be calculated for any given value of the ratio D/d. Adding this stress to the tensile stress p_1 produced by the longi-

TABLE 1

$\frac{D}{d}$	3	4	5	6	8	10
$\frac{2\rho}{h}$	0.1706	0.2228	0.2574	0.2838	0.3239	0.3536
$\frac{p_1}{p_0}$	1.50	1.33	1.25	1.20	1.14	1.11
$\frac{p_2}{p_0}$	2.33	1.93	1.83	1.83	1.95	2.19
$\frac{p_{max}}{p_0}$	3.83	3.26	3.08	3.03	3.09	3.30

tudinal force S_0 (see Equation [3]), the complete stress at the point n_0, which represents the maximum stress, will be obtained in the following form:

$$p_{max} = p_1 + p_2$$

The results of calculations made in such a way for different values of the ratio D/d are given in the Table 1. The corresponding stress

distribution over the cross-section through the hole is shown in Fig. 1(*a*) by the shaded area.

11 The bottom row of values in Table 1 represents the ratio of maximum stress at the edge of the hole to the uniform tensile stress p_0 applied at the end cross-section of the plate (see Fig. 1). From now on this ratio will be called the "factor of stress concentration" and will be denoted by k. Therefore

$$p_{max} = kp_0 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [11]$$

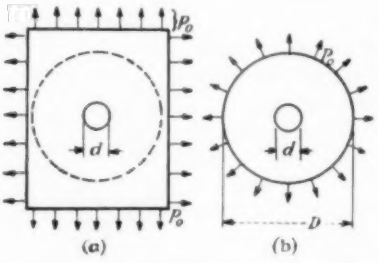


FIG. 3 PLATE WITH CIRCULAR HOLE WHERE EQUAL TENSILE OR COMPRESSIVE STRESSES ARE ACTING IN TWO PERPENDICULAR DIRECTIONS

12 Comparing the data given in Table 1 with the exact solution, $k = 3$, for $D/d = \infty$, it is seen that for the ratios lying within the limits $5 < D/d < 8$ the results obtained in this elementary manner agree closely with exact solution for a small circular hole. When $D/d < 5$, the hole produces a perceptible weakening of the cross-section, and due to this fact the factor of stress concentration becomes larger than 3. The increase of the ratio p_{max}/p_0 in the cases when $D/d > 8$ is the result of insufficient accuracy in the elementary theory of curved bars when the inner radius of the ring is very small in comparison with the outer. This comparison of the elementary method of calculating stress concentration with the exact theory shows that the approximate solution is accurate enough for practical applications and may be used in cases where an exact solution is not available. One example of this kind will be considered below.

III — COMBINED TENSION OR COMPRESSION IN TWO PERPENDICULAR DIRECTIONS

13 If equal tensile or compressive stresses are acting in two perpendicular directions (Fig. 3), the stress concentration at the edge of a circular hole can be found by using Lamé's formula for thick cylinders. [Fig. 3(*b*)]. On the basis of this theory the tensile stress in a tangential direction at the edge of the hole becomes

$$p_{max} = 2p_0 \frac{D^2}{D^2 - d^2}$$

and for a small hole

$$p_{max} = 2p_0 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [12]$$

This means that due to a small circular hole the stress in the case under consideration will be doubled.

14 Comparing Equation [**12**] with the equation

$$p_{max} = 3p_0 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [13]$$

obtained for simple tension, it can be concluded that for the case of simple tension a compressive stress p_{min} determined by the equation

$$p_{min} = -p_0 \quad . \quad . \quad . \quad . \quad . \quad [14]$$

acts in a tangential direction at points a, Fig. 4. Then by using the principle of superposition we obtain Equation [**12**] from Equations [**13**] and [**14**].

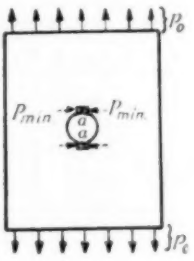


FIG. 4

15 The same principle of superposition gives the solution of the problem presented in Fig. 5. By using Equations [**13**] and [**14**] it can be shown that the tangential stress at the points b at the edge of the hole is equal to $3p_0 - q_0$. The same stress for the points a is equal to $3q_0 - p_0$. When $q_0 = -p_0$, i.e., in the case of "pure shear," we have for the points b,

$$p_{max} = 3p_0 - (-p_0) = 4p_0 \quad . \quad . \quad [15]$$

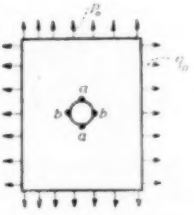


FIG. 5

16 For the points a,

$$p_{min} = 3q_0 - p_0 = -4p_0$$

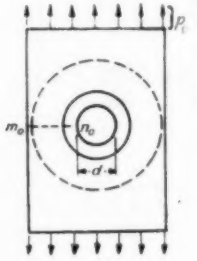
This means that in the case of twisting a thin circular tube (Fig. 6) a small circular hole produces a very high stress concentration, such that the maximum tensile stress at the edge of the hole becomes four times as large as the shearing stress uniformly distributed over the ends of the tube. This conclusion will be accurate enough also in the case of a solid shaft under torsion having circular holes in a radial direction, such as oil ducts in crankshafts.

FIG. 6

IV — CIRCULAR HOLE WITH BEAD

17 In order to diminish the weakening effect of a hole it is a common practice to reinforce its edge with a bead as in Fig. 7. It may be expected that the bead will diminish the value of the maximum stress and also the region of the perceptible local stresses due to the hole. Under such conditions the approximate method, developed above (see Sec. II), will be more satisfactory than for

a non-reinforced section. By applying this method, the problem of stress concentration again will be reduced to the calculation of stresses in a circular ring with an outer diameter D and inner diameter d (Fig. 7). In calculating the bending moment M_0 on the cross-section m_0n_0 of this ring the general Equation [4] can be used. Denoting by c_1 and c_2 the distances of the center of gravity of the cross-section (Fig. 8) from the most remote fibers and letting



$$\frac{b}{t} = n \qquad \frac{a}{d} = m \quad ,$$

we have

$$\frac{c_1}{d} = \frac{h}{2d} \cdot \frac{1 + (n-1)\dfrac{m^2d^2}{h^2}}{1 + (n-1)\dfrac{md}{h}} \quad . \quad . \quad [16]$$

18. If m is a small quantity, Equation [16] can be represented in the following simplified form:

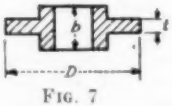


FIG. 7

$$\frac{c_1}{d} = \frac{h}{2d}\left(1 - (n-1)\frac{md}{h}\right) \quad . \quad . \quad [17]$$

The distance γ of the neutral axis from the center of gravity of the cross-section will be found in the usual manner from the equation

$$\frac{\gamma}{d} = \frac{\rho - r}{d} = \frac{\rho}{d} - \frac{(n-1)m + \dfrac{h}{d}}{(n-1)\log_e(1+2m) + \log_e \dfrac{D}{d}} \quad . \quad . \quad [18]$$

19. When m is a small quantity, the following approximate equation can be used instead of Equation [18]:

$$\frac{\gamma}{d} = \frac{\rho}{d} - \frac{(n-1)m + \dfrac{h}{d}}{2m(n-1) + \log_e \dfrac{D}{d}} \quad . \quad . \quad . \quad . \quad . \quad [19]$$

The quantities m and $n-1$ enter into Equations [17] and [19] as the product term $m(n-1)$ only. Remembering that

$$m(n-1) = \frac{a(b-t)}{dt}$$

and letting

$A_b = a(b-t)$ = area of the cross-section of the bead
$A_h = dt$ = area of the diametrical section of the hole

it can be concluded that the positions of the center of gravity and neutral axis depend on the magnitude of the ratio

$$\alpha = m(n-1) = \frac{A_b}{A_h} \quad \ldots\ldots\ldots \quad [20]$$

This ratio will be used in further calculations.

20 After calculating c_1, c_2, and γ from the equations given above, the magnitude of the bending moment M_0 on the cross-section m_0n_0 (Fig. 7) will be found from Equation [7]. It is only necessary to put in this equation the distance c_2 of the center of gravity of the cross-section from the outer edge of the ring, instead of $h/2$, and to take an appropriate expression for γ.

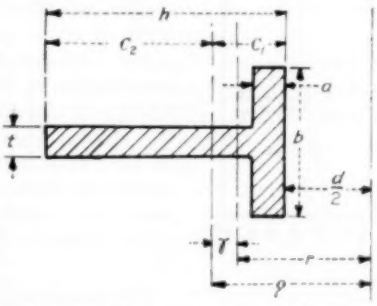


FIG. 8 CROSS-SECTION FOR CASE OF SMALL CIRCULAR HOLE REINFORCED WITH A BEAD

21 Denoting by A the area of the cross-section of the ring,

$$A = th + a(b-t)$$

the tensile stress produced in the cross-section m_0n_0 by the longitudinal force will be

$$p_1 = \tfrac{1}{2}\,\frac{p_0 lD}{A} \quad \ldots\ldots\ldots \quad [21]$$

22 The bending stresses in the same cross-section will be found from the general Equation [9]. It is necessary only to put $M = M_0$ and $S = A\gamma$. The maximum bending stress will be in the point

TABLE 2

$\alpha = \frac{A_b}{A_h}$	$\frac{c_1}{d}$	$\frac{\rho}{d}$	$\frac{\gamma}{d}$	$\frac{c_2}{d}$	$\frac{p_1}{p_0}$	$\frac{p_2}{p_0}$	$k_1 = \frac{p_1+p_2}{p_0}$
0.10	0.953	1.453	0.291	1.047	1.19	1.34	2.53
0.20	0.9095	1.4095	0.312	1.0905	1.14	1.03	2.17
0.30	0.8703	1.3702	0.326	1.1297	1.09	0.81	1.90
0.40	0.8342	1.3342	0.334	1.1658	1.04	0.65	1.69
0.50	0.8010	1.3010	0.339	1.1990	1.00	0.53	1.53

n_0 at the edge of the hole. For this point $z = c_1 - \gamma$, and $r - z = r + \gamma - c_1 = \frac{1}{2}d$, and Equation [9] gives

$$p_2 = \frac{M_0}{A\gamma}\,\frac{2(c_1-\gamma)}{d} \quad \ldots\ldots\ldots \quad [22]$$

The maximum stress will now be found from the equation

$$p_{max} = p_1 + p_2 = k_1 p_0 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [23]$$

23 In order to show what effect the increasing of the area of the cross-section of the bead has on the magnitude of the *factor of stress concentration*, calculations were made for several values of the ratio $\alpha = A_b/A_h$. The results of these calculations are given in Table 2.[1]

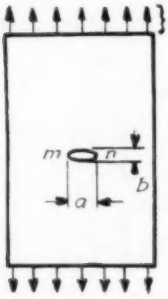


FIG. 9

24 The last column of Table 2 gives the magnitudes of the factors of stress concentration k_1 for different values of the ratio $\alpha = A_b/A_h$. It is seen that as the cross-section of the bead increases the magnitude of the maximum stress decreases. For $\alpha = 0.50$ this maximum is about one-half that in the case of a hole without a bead.

V — Elliptical Hole[2]

25 If a plate subjected to a uniform tension has a small elliptical hole (Fig. 9), the larger axis of which is perpendicular to the direction of tension, very high stress concentration takes place. The maximum tensile stress occurs at points m and n where the curvature of the edge of the hole is the greatest. The magnitude of the maximum stress is given by the equation

$$p_{max} = p_o\left(1 + \frac{2a}{b}\right) . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [24]$$

26 When $a = b$, i.e., for a circular hole, Equation [24] reduces to Equation [13]. As the ratio a/b increases, the factor of stress concentration increases also and becomes, in the case of a narrow elliptical hole, very high. This explains in a measure why cracks extending perpendicularly to the direction of stress are likely to spread very quickly and why this spreading may be stopped by drilling holes at the ends of the crack. Such holes reduce the curvature and diminish substantially the factor of stress concentration. The same Equation [24] may be used also in the case when the larger axis of the elliptical hole is parallel to the direction of tension. In this case the ratio a/b will be less than unity and the factor of stress concentration will be less than for a circular hole.

[1] In these calculations it was assumed that $D/d = 5$. Preliminary calculations showed that the stress concentration remains practically constant when this ratio varies within the limits $4 < D/d < 6$. See paper by S. Timoshenko in *Journal of The Franklin Institute*, April, 1924.

[2] This problem was solved first by G. Kolosoff. See his thesis, University of St. Petersburg (1909). See also C. E. Inglis' paper in Proc. Inst. of Naval Architects, London, March 14, 1913.

VI — Photoelasticity Applied to the Fillet Problem

27 *General.* Stress concentrations may be determined by the photoelastic method. In brief, this method consists of loading models made from isotropic material such as celluloid and subjecting them to polarized light. The degree of birefraction is a function of the loading, and a definite color pattern is obtained for a given condition of stress. The interpretation of this color pattern makes possible the evaluation of these stress intensities. It must be borne in mind that all values obtained and conclusions drawn are only applicable so long as the problem is two-dimensional and Hooke's law is obeyed.

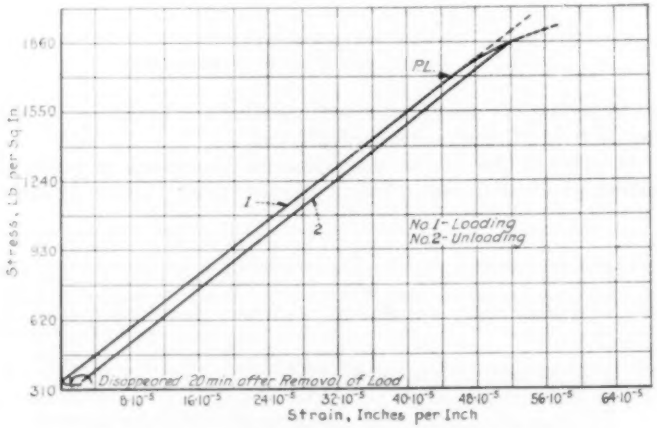


Fig. 10 Stress-Strain Diagrams for Celluloid

(E=302,000 lb. per sq. in.; proportional limit, 1700 lb. per sq. in.; Poisson's ratio, 0.37.)

28 In the work described in the following paragraphs, celluloid obtained from the DuPont de Nemours & Co. was used which was $\frac{1}{8}$ in. thick. The stress-strain relation obtained is shown by the curve Fig. 10, from which the proportional limit was found to be 1700 lb. per sq. in. and Young's modulus 302,000 lb. per sq. in. Poisson's ratio for this material was 0.37.

29 *Fillets in Tension.* The models used were of the type shown in Fig. 11 and fillets having radii ranging from $\frac{1}{2}$ in. to 1/16 in. were studied. The variations in the width of the model were expressed by the ratio D/d, Fig. 11. The ratio D/d in all models for the first series of measurements was equal to 6 ($D/d = 6$). After finishing this series the models were then symmetrically machined on the width D so as to bring the ratio D/d to approximately 1.1 through

stages of $D/d = 3.5, 2.5, 2, 1.5$, etc. For each stage a complete series of stress measurements were taken at arbitrarily chosen points on the periphery as shown in Fig. 15(*a*). Point *A* was 0.5 in. from start of fillet, *B* at start of fillet, *C* at a distance from

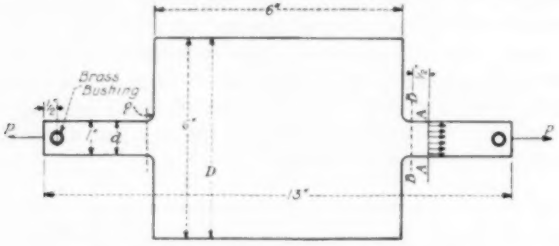


FIG. 11 MODEL FOR FILLET INVESTIGATION

B subtending $11\frac{1}{4}$ deg., *D* at a distance from *C* subtending $11\frac{1}{4}$ deg. and *E*, *F* and *G* each subtended 22.5 deg.

30 The results of the measurements are given in terms of factors of stress concentration. By the term "factor of stress

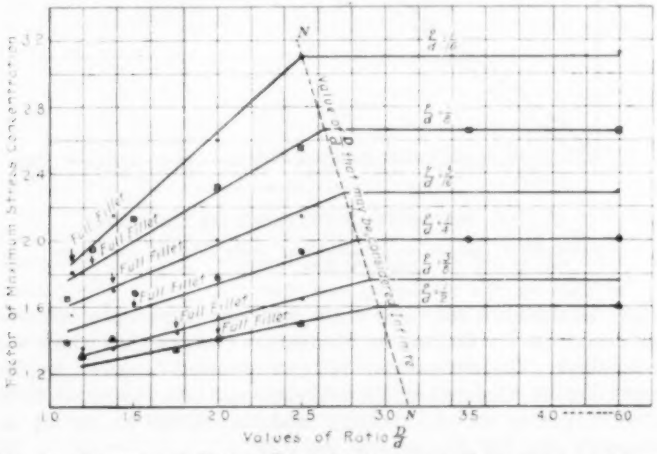


FIG. 12 CURVES FOR STRESS CONCENTRATION VS. REDUCTION OF CROSS SECTION

(*D*=width of model; *d*=width of approach; *ρ*=radius of fillet—see Fig. 11.)

concentration" is meant in this case the ratio of a maximum stress at a given point on the fillet to the stress obtained in the approach on the assumption of a uniform tension. The maximum stress at the point is the stress which is tangent to the periphery at that point; that is, it is one of the principal stresses. From the optical

measurement of the stress distribution across the approach, it followed within the limit of accuracy of the experiment that in the cross-section A, Fig. 11, $\frac{1}{2}$ in. distant from the start of the fillet, the *stress distribution was uniform*. This stress, as measured optically, agreed closely with calculated average stress, which latter value was used in obtaining the factors of stress concentration in tension.

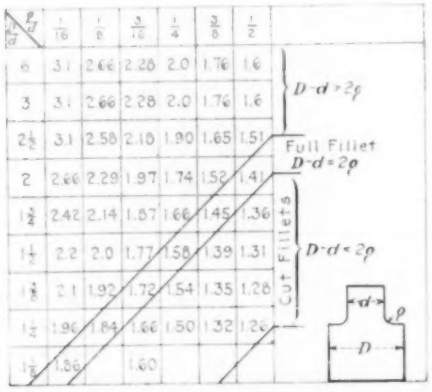


FIG. 13 FACTORS OF MAXIMUM STRESS CONCENTRATION FOR FILLETS

This table gives the factors of maximum stress concentration by fillets of different radii and varying conditions of areas reduced. Linear interpolation may be used throughout.

31 The complete data of the experiments are given in Fig. 12, in which all points obtained are plotted. It might be mentioned in advance that it is logical to expect that above a certain value of the ratio D/d, the variation in the width D will not materially affect the maximum stress concentration at the fillet. It is probable that the limiting value of D/d depends upon the magnitude of the radius of the fillet, that is, that it increases as ρ/d increases. The experimental data confirm these expectations and it is seen from Fig. 12 that the results for the larger values of D/d can be very satisfactorily approximated by straight lines parallel to the horizontal axis. It is seen that for $\rho/d = \frac{1}{2}$ the limiting value of D/d can be taken equal to 3. This means that for all values of $D/d > 3$ the stress distribution at the fillet remains practically the same. For another extreme case, $\rho/d = \frac{1}{16}$, the limiting value of D/d is about 2.5. The limiting value of D/d for intermediate values of ρ/d will be determined by the straight line $N-N$ shown in Fig. 12. For values of D/d smaller than those given by the line $N-N$ the coefficients of stress concentration diminish with decrease of D and may be represented with sufficient accuracy by the inclined lines shown in Fig. 12.[1]

[1] These lines should not be extrapolated for the region D/d close to 1.

32 From the system of lines in Fig. 12 the table of factors of maximum stress concentration given in Fig. 13 was prepared.

33 It was mentioned above that if $D/d > 3$ the factor of maximum stress concentration, practically speaking, does not depend upon the value of D/d and varies only with a variation in the ratio ρ/d. This case has been dealt with separately and the results plotted in Fig. 14. The curve LM represents the experimental data. It is seen that over a comparatively large range this experimental curve is in very good agreement with the broken-line hyperbola, the equation of which is $x(y-1.3) = 0.172$, in which x equals ρ/d and y the factor of maximum stress concentration.

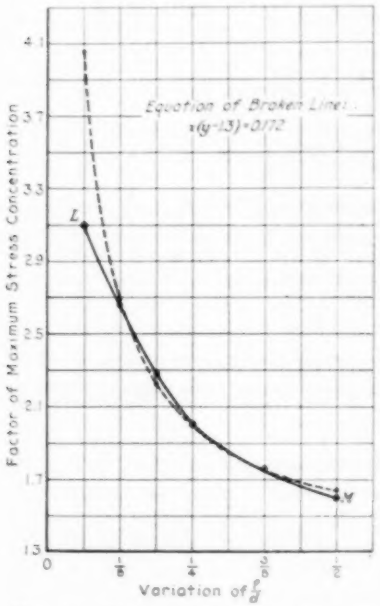


FIG. 14 STRESS CONCENTRATION VS. FILLETS WITH $D/d = 6$

34 A study of the stress distribution throughout the entire fillet was made for the purpose of determining the variations with different values of D/d. In Fig. 15 are plotted curves for the fillet in which ρ/d is equal to $\frac{1}{8}$. The ratio of D/d was taken equal to 6, $1\frac{1}{4}$, and $1\frac{1}{8}$, respectively. It is to be noted that the maximum stress concentration shifts toward the start of the fillet with decreasing values of D/d. For $D/d = 1\frac{1}{8}$, which corresponds to the fillet being cut in half, the point of maximum stress is at the beginning of the fillet.

35 *Fillets in Bending.* The following experiments with fillets in bending were carried out to determine whether the table given in Fig. 13 and obtained with fillets in tension could be applied for fillets in bending. The type of model used and the method of loading are shown in Fig. 16. It is seen that the stress distribution in the approach is quite different from that of the previous experiment, Fig. 11, one fillet being in tension and the other in compression. As a reference for calculating the coefficient of stress concentration it was decided to use the stress that would be present at the point corresponding to the start of the fillet if there were no fillet effect. To obtain this the following procedure was adopted. A point A', Fig. 16, was taken on the border of the approach

midway between the start of the fillet and the application of the load. The stress distribution at this section was given accurately enough by the standard formula $f = MY/I$ and shown in Fig. 16. The calculated value of f checked the photoelastic values of f_0 at A' on the border of the approach within the limit of accuracy of the experiment. Having the stress at A', the stress at B' must be twice that at A' on the assumption of no fillet effect. Therefore twice the calculated value at A' was used as a reference for calculating the factor of stress concentration. Fillets of two sizes were used, $\rho/d = \frac{1}{4}$ and $\rho/d = \frac{7}{16}$. The ratio of D/d was kept equal to 6. The position of maximum stress was the same as in the previous experiments, approximately 22.5 deg. along the fillet. The results obtained are given in Table 3.

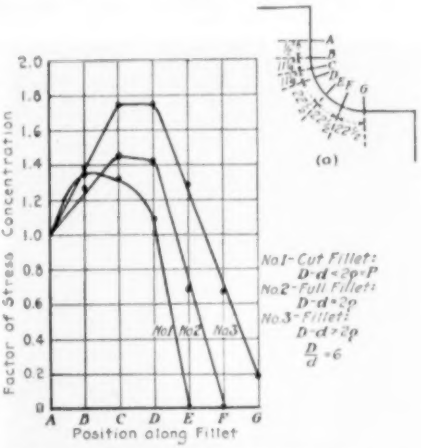


FIG. 15 STRESS CONCENTRATION VS. POSITION ON FILLET ($\rho/d = \frac{7}{16}$)

36 It is seen that the values in Table 3 are in good agreement with the corresponding values given in the table of Fig. 13, and it was concluded that the latter holds also for fillets in bending.

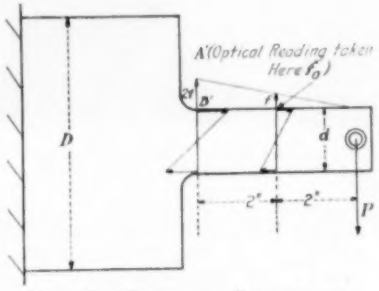


FIG. 16 MODEL AND LOADING FOR FILLET IN BENDING

37 The values given in these two tables are not in error more than ±5 per cent. This limit for the error was obtained by considering the various possible sources of error throughout the work.

38 *Conclusions.* The experiments made show that there is a high stress concentration produced by fillets, the magnitude depending principally on the ratio of the radius of the fillet to the width of the approach. The experimental results obtained for the complete problem of fillets in tension and in bending are given in Fig. 13.

VII — Lueders' Line Method Applied to Problems of Fillets and Holes

39 If a mild-steel tensile-test specimen be highly polished and subjected to an increasing load, very definite inclined lines will appear on the surface as the load approaches the yield point. These lines, known as "Lueders' lines," are well defined (see Fig. 17) and can be used to establish the regions of highest stress concentration in a model of any machine part as well as the approximate magnitude of the factor of this concentration. To establish this magnitude the following procedure is adopted. An ordinary tensile test of a polished specimen is made to ascertain the stress at which the "lines" appear for a particular material. Having this datum, the further experiment with models can be made. The models of the same material are then loaded until the Lueders' lines become visible in the overstressed regions. They indicate in this manner the points of highest stress concentration. In order to obtain the magnitude of the factor of stress concentration, the "limiting stress" for the simple test bar is divided by the "limiting stress" for the model. The limiting stress is that value of stress which just causes lines to appear. For the fillet model this is calculated by dividing the load corresponding to the appearance of the lines by the area of cross-section at the approach (see Fig.

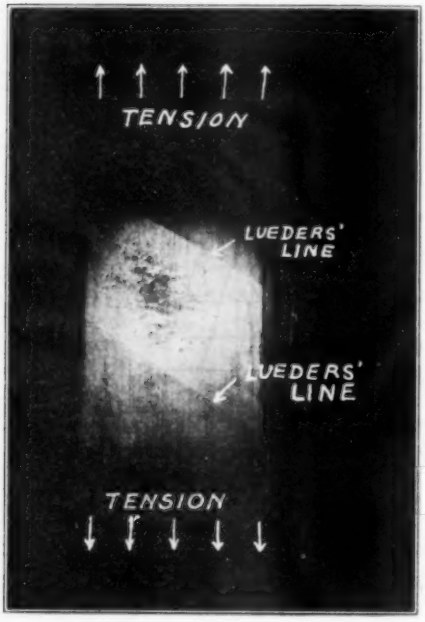


Fig. 17 Lueders' Lines for Case of Simple Tension

TABLE 3

$\frac{p}{d}$	Condition	Factor of Stress Concentration
$\frac{1}{8}$	Bending (tension)	2.02
$\frac{1}{8}$	Bending (compression)	2.10
$\frac{1}{4}$	Bending (tension)	1.66
$\frac{1}{4}$	Bending (compression)	1.64

11). For the case of a plate with a circular hole the area of unweakened section must be used in calculating the limiting stress.

40 Fig. 18 shows a Lueders' line on a fillet model of mild steel having the proportions $\rho/d = 0.157$ and $D/d = 2.5$. The line started from the point on the fillet where, on the basis of the previous photoelastic experiments, the maximum stress concentration was expected. The factor of stress concentration calculated, as explained above, is as high as 1.8. The corresponding value of the same factor obtained by photoelastic method is 2.4. In the case of a circular hole (Fig. 19) the Lueders' lines again show accurately the points of maximum stress concentration. The factor of stress concentration as determined by the "lines" method is 2.3 as against 3 calculated analytically above. In both cases the values of the factor of stress concentration obtained by Lueders' lines are approximately 25 per cent lower than those given by the photoelastic method. This is mainly due to the fact that the region of stress concentration is very small and it is very difficult to observe the lines at the moment they are formed. Considerable time therefore elapses from their beginning before they become clearly visible.[1] A method for better condition of observance of the lines is being developed which it is hoped will greatly reduce the discrepancy now encountered. This method of testing the weak points of structures is not confined to any particular type of problem and has an advantage over the photoelastic method in that it is applicable to three-dimensional problems.

FIG. 18 LUEDERS' LINES ON A FILLET MODEL OF MILD STEEL

[1] The yielding of the material at the points of maximum stress must also have some effect on the retardation of lines.

VIII — Measuring of Strain Produced by Stress Concentration

41 Due to the stress concentration occurring at certain points on the edges of holes or fillets and which has been previously discussed, large local strains also take place at these points. There strains can be measured by using fine extensometers with a small

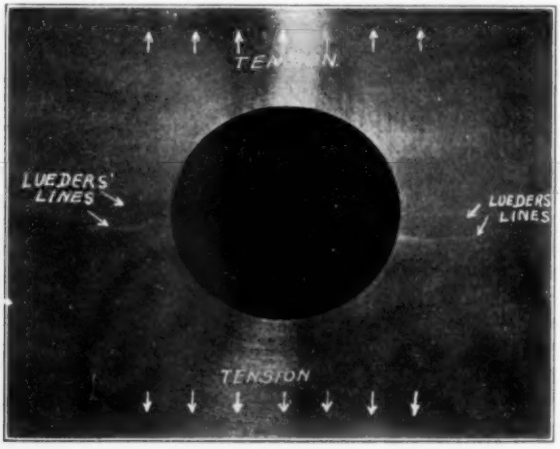


FIG. 19 LUEDERS' LINES IN CASE OF A CIRCULAR HOLE

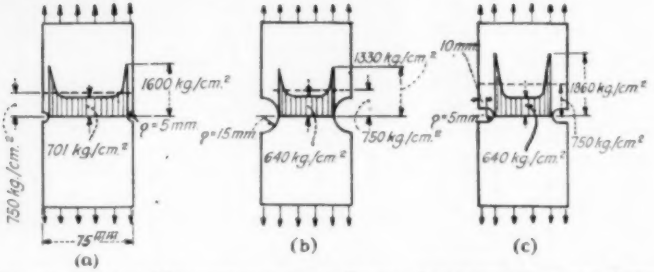


FIG. 20 STRAIN MEASUREMENTS FOR STRESS CONCENTRATION FOR STEEL PLATES WITH DIFFERENT KINDS OF NOTCHES

gage length. Many experiments of this kind were made by Professor Preuss.[1] Some of the results obtained in this manner for steel plates with different kinds of notches are shown in Fig. 20. The tensile force in all cases was such that the average stress over the weakest cross-section was equal to 750 kg. per sq. cm. The distribution of tensile stresses is shown in the figures by the shaded areas.

[1] See Professor Preuss's papers in *Zeit. d. Ver. deutscher Ing.*, 1912, p. 1349, and 1913, p. 664.

42 The results obtained for semicircular notches are in good agreement with the calculations made for this case.[1]

IX — Practical Limitations

43 All the data on stress concentration obtained by analytical methods or by using photoelastic experiments are based on the assumption that the stress distribution is a two-dimensional one and that the material is homogeneous, isotropic, and perfectly elastic. Such material as steel satisfies these conditions with good accuracy within the limit of elasticity and the data given in the paper represent sufficiently accurately the actual stress concentrations produced by holes, notches, or fillets for the cases of two-dimensional problems. While these data have been specifically obtained on two-dimensional models, they can assist also in the more complicated cases of three-dimensional stress distribution. Take, for instance, the case of a crankpin with circular holes a in a radial direction (Fig. 21). On the basis of the solution that has been given for a circular hole, it can be concluded that the maximum stress at the edges of the holes in the crankpin will be about three times greater than it is given by the usual bending-stresses formula.

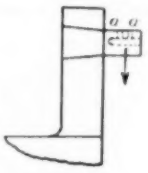


Fig. 21

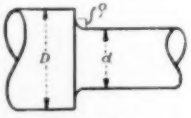


Fig. 22

44 In the case of a circular shaft having a rapid change in section (Fig. 22) the maximum bending stress at the fillet will be much greater than is given by the usual theory of bending. Some idea of the magnitude of the factor of stress concentration can be secured from the Table 3, the values in which were obtained by the photoelastic method. It is reasonable to believe that if the magnitude of the ratio ρ/d is small, say, $\rho/d < \frac{1}{12}$, the results given in Table 3 represent approximately the fillet problem for a circular shaft. It is easy to see that with a decrease in the ratio ρ/d the factor of stress concentration increases, and in the case of a sharp corner, Fig. 23(a), this factor becomes, theoretically speaking, infinitely large. This means that at a sharp reëntrant corner a small bending moment will produce a permanent set. This plastic deformation is of highly localized type and will not affect the deflection curve of the shaft. In order to remove this stress concentration a semicircular groove, as shown in Fig. 23(b), is sometimes used.

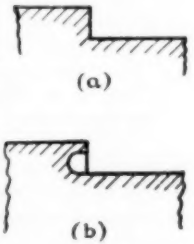


Fig. 23

[1] A. Leon, Mitteilungen Mech. Techn. Labor., Wien, 1908.

45 In considering the weakening effect of stress concentration on the ultimate strength of the structure submitted to a static loading, it is necessary to keep in mind that the factors of stress concentration, given above for several cases are true only within the elastic limit of the material. Beyond this limit the effect of stress concentration depends principally on the ductility of the material. For ductile materials, as for mild steel, with an increase of loading beyond the elastic limit the local stresses in the overstressed region will not increase in the same proportion due to yielding of material, and in the cases of tensile tests of bars with

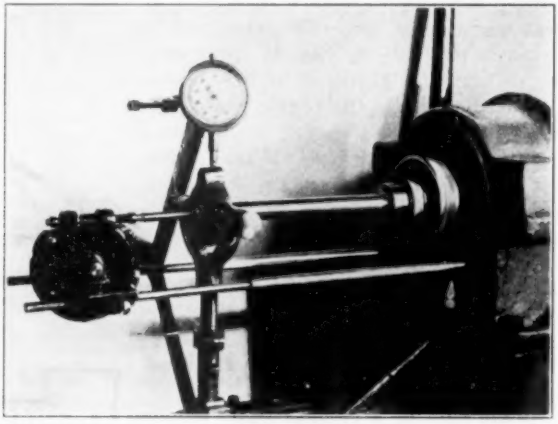


FIG. 24 CANTILEVER FATIGUE-TESTING MACHINE IN WHICH REVERSAL OF STRESS IS PRODUCED IN THE ROTATING SPECIMEN BY CONSTANT VERTICAL LOAD

holes or notches the stress distribution over the weakened cross-section at the moment of break can be considered as uniform. This means that stress concentration will have no effect on the ultimate strength of the bar. In some cases an increase of ultimate strength may be expected due to the fact that the lateral contraction at the weakened cross-section will be prevented to some extent.[1]

46 Another result will be obtained in the case of a brittle material. In this case plastic deformation, allowing equalization of stresses over the weakened cross-section, does not occur and the break will start from the points of highest concentration of stress. For such materials as glass or hardened steel, holes and notches will diminish the ultimate strength in a ratio which agrees approximately with the factors of stress concentration calculated or obtained by the photoelastic method.

47 The most important effect of stress concentration obtains in the cases where a reversal of stresses takes place. In such cases

[1] See C. Bach, Elasticität und Festigkeit, 1920, p. 153.

the stress concentration produces a weakening effect not only on brittle but also on ductile materials. Due to reversal of stress at points of high local stresses the development of a crack begins, which gradually spreads over the weakened cross-section of the structure and finally produces failure. This occurs at a loading much lower than could be expected on the basis of the usual theory. In order to procure some data for such conditions, several fatigue tests were made by using a cantilever fatigue-testing machine in which the reversal of stress is produced in the rotating specimen by a constant vertical load. (See Fig. 24). Cylindrical specimens with the proportions $\rho/d = \frac{1}{12}$ and $D/d = 2$ were used for this purpose. (Fig. 25.) The value of a factor of stress concentration for this case, as obtained from Table 3, is about 2.5. The physical properties of two materials tested are given in Table 4.

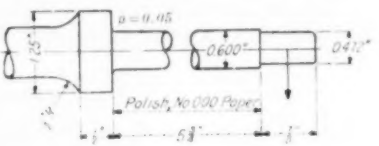


FIG. 25 CYLINDRICAL SPECIMENS USED IN FATIGUE TESTS

48 In column 7 of Table 4 are given the endurance limits which are obtained from the tests of standard fatigue-test specimens where the action of stress concentration is excluded. In column 8 the endurance limits are given for specimens with fillets (Fig. 24).[1] The ratio n of the figures given in columns 7 and 8 represents the weakening effect of the fillets. This effect, as we see, is sufficiently

TABLE 4

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
Material	Proportional limit, lb. per sq. in.	Yield point, lb. per sq. in.	Ultimate strength, lb. per sq. in.	Elongation, per cent	Reduction in area, per cent	Endurance limit, lb. per sq. in.	Reduced endurance limit, lb. per sq. in.	n (col. 7 ÷ col. 8)
Carbon steel...	41,000	41,000	73,000	35	57	32,000	21,000	1.5
Ni-Cr steel.....	73,000	75,000	106,000	27	61	52,000	31,000	1.7

large and cannot be neglected in actual design. It is conceivable that the ratio of the factor n obtained from fatigue tests to the factors of stress concentration given by the table of Fig. 13 depends principally on the ductility of material. For brittle materials this ratio must approach the value unity. For the nickel-chromium steel given in Table 4 the effect of stress concentration is about half what could be expected on the basis of the data given in Fig. 13.

49 In considering a stress concentration produced by holes and grooves it has been assumed in theoretical discussion that the dimensions of these holes and grooves are small in comparison with

[1] Calculated on the basis of a simple bending formula.

the dimensions of the cross-section of bars under consideration. If this condition is satisfied the factor of stress concentration will not depend on the size of the hole, but the weakening effect of this stress concentration in the case of very small holes or fine scratches on the surface depends, as it is now shown by experiments,[1] on the linear dimensions of the region of high stress. In one case of hardened cast steel[2] with scratches having a depth of the order of 10^{-4} in., no weakening effect of stress concentration was observed. But with the same material, having grooves 0.025 in. deep, the greater part of the theoretical weakening was developed. On basis of this *scale effect* it can be explained why scratches corresponding to different kinds of finishing of the surface of fatigue-test specimens have not a great weakening effect on the endurance limit of material.

X — Conclusions

50 On the basis of the analytical solutions and experimental data given above, it can be concluded that in many practical cases a very high stress concentration is produced by holes, grooves, and sharp variation of cross-sections. In the case of ductile materials this stress concentration does not have a weakening effect under static loading. In the cases of brittle materials or ductile materials under the action of stress reversal, however, the weakening effect of stress concentration may become of prime importance and it must be taken into consideration in actual design.

51 In the case of brittle materials the weakening effect of stress concentration is given with a good approximation by the factor of stress concentration which can be established analytically or experimentally by one of the methods that have been described.

52 The weakening effect of stress concentration in the case of ductile material submitted to the action of stress reversal can be established accurately enough only on the basis of fatigue tests of the models of the machine parts under consideration. Tests of this kind are now in progress in the research laboratory of the Westinghouse Electric & Manufacturing Company.

DISCUSSION

H. F. Moore.[3] In this valuable paper there have been given the results of some repeated-stress tests of steel specimens showing that as regards fatigue strength the weakening effect of stress concentration at fillets is about one-half the effect called for by the results of photoelastic tests on celluloid specimens. Supplementary

[1] See A. A. Griffith, Phil. Trans. A, vol. 221, pp. 163-198, 1920; and Proc. British Assoc. Adv. Science, 1922.

[2] Experiments of A. A. Griffith, cited above.

[3] Research Professor of Engineering Materials, University of Illinois, Urbana, Ill. Mem. A.S.M.E.

to this result, and in general confirming it, the writer wishes to present the results of some fatigue tests at the University of Illinois.

These tests were on specimens of ten kinds of steel. The specimens were flat specimens pierced with a hole, as shown in Fig. 26. The fatigue strength determined for each kind of steel from tests of such flat specimens was compared with the endurance limit determined from tests of rotating-beam specimens of the same steel, in which there was very little concentration of stress due to form of specimen. The general results of these tests showed that at the hole in a flat specimen there was an effective stress concentration factor (using the term and definition given by the authors' paper) varying from 1.25 to 1.75 for different kinds of steel, with an average of 1.35. From either the Inglis formulas for stress at holes, or from the authors' approximate treatment the stress concentration factor is found to be about 3.

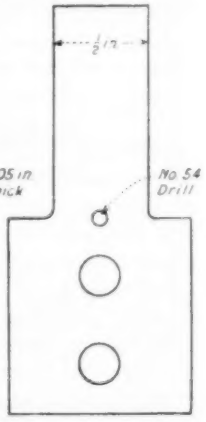


FIG. 26 TEST SPECIMEN FOR FATIGUE TESTS

Another method of studying stress concentration has recently been tried out in a tentative way by R. E. Peterson, a senior student in mechanical engineering at the University of Illinois, working under the immediate direction of the writer. This method involves the use of specimens of very brittle material [1] and the material used by Mr. Peterson was dental plaster which, properly mixed and hardened for about a week, has a stress-strain diagram in tension which is very nearly a straight line up to rupture. Mr. Peterson made tests of rectangular beams with various sized holes. A test beam is shown in Fig. 27. After a test beam had been broken through the hole the halves were again broken, giving a comparison between a beam with stress concentration due to a hole and one without such stress concentration. Mr. Peterson's test results are shown in Fig. 28.

The use of plaster specimens for studying stress concentration or for other problems of stress determination is offered as a less elegant method than the use of polarized light with celluloid specimens, but an inexpensive one which can be used on "three-dimensional" specimens, and without elaborate apparatus. It is to be hoped that the possibilities and limitations of this method may be further investigated.

[1] See Bach, Elasticität und Festigkeit, eighth edition (1920), Chap. V, especially the portion beginning at page 365; also Kommers Torsion Tests of Cast Iron, *American Machinist*, May 28, 1914, p. 941, for previous use of this general method of stress determination.

It is to be noted that the plaster-beam tests, the fatigue tests of steel specimens, and the Lueders' line tests used by Dr. Timoshenko all give values of stress concentration factor less than the

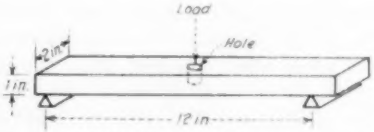


FIG. 27 TEST BEAM OF DENTAL PLASTER

values called for by the theory of elasticity, and also less than the values given by photoelastic tests on celluloid specimens. The following is offered as a partial explanation of this discrepancy.

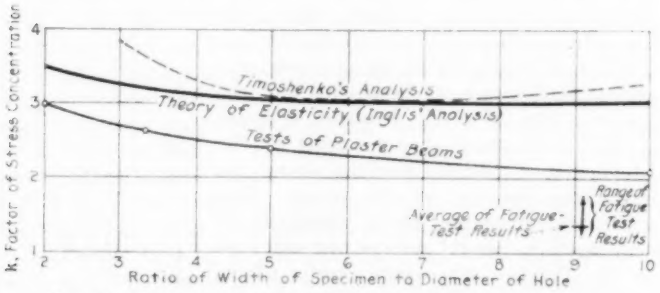


FIG. 28 RESULTS OF TESTS ON BEAM OF DENTAL PLASTER

The theory of elasticity gives a value of a stress or strain at any given point, and if the value of maximum stress or maximum

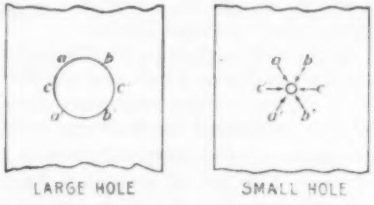


FIG. 29 EFFECT OF STRESS ON PLATES WITH LARGE AND SMALL HOLES (*cc'*, points of maximum stress intensification; *ab* and *a'b'*, regions of understressed material which act to restrain lateral deformation at *cc'*.)

strain is used as a criterion of the effective structural damage to a machine part there is implied the assumption that whenever a certain critical stress intensity is reached at any point whatever

the machine part at once fails. This assumption completely neglects the fact of ductility in materials. All materials possess some degree of ductility, even such brittle materials as cast iron, concrete, and dental plaster. Under static load, ductility is very effective in preventing a high localized stress in a member from causing actual rupture. Stretch, or compression, or shearing detrusion occurs with an accompanying redistribution of stress and a tendency to diminish the stress at its point of maximum intensification. So effective is ductility in this respect that in static tests to failure of ductile metals, holes and fillets have practically no effect on the ultimate tensile strength of specimens. The ameliorating effect of ductility is much less for brittle materials than for ductile materials, but even for brittle materials it seems reasonable to suppose that the slight degree of ductility present diminishes somewhat the destructive effect of localized intensified stress. The ameliorating effect of ductility is less for repeated loading than it is for static loading, but it seems to be of some effect even under repeated loading.

In conversation with the writer recently Prof. F. B. Seely of the University of Illinois mentioned another effect which may be of importance in the case of brittle materials. From a comparison of test results of short compression specimens of brick and concrete with test results for medium length specimens it seems evident that for short specimens the lateral expansion which accompanies longitudinal compression is restrained from attaining its full value in short specimens. Analogous restraint of lateral deformation is found in the case of screw threads under tension, and both for tension tests and for compression tests this restraint of lateral deformation seems to increase the strength of the specimen. At points of localized stress such lateral restraint would be present in a greater or less degree, and for the case of holes in a specimen the smaller the hole the closer at hand is understressed material which can act to set up such restraint (see Fig. 29). This is offered as an explanation of the observed result that for small holes the discrepancy between results computed from the theory and test results on specimens is greater than is the case for large holes.

A general result, which is of especial interest to the designer and which stands out from all available test results, is this: The use of the formulas of the theory of elasticity for stress intensification, the use of the authors' formula for holes, and the use of photoelastic tests on transparent specimens, all give results which are on the safe side as judged by the results of static and fatigue tests.

T. McLEAN JASPER.[1] The writer desires to present certain experimental results bearing on the matter of stress concentration and its effects, particularly on steel shapes under reversed stress tests.

The authors in their foreword have referred to the fact that "When reversal of stress is involved the weakening effect of stress concentration can only be established accurately on the basis of fatigue tests." This discussion, therefore, will bear mainly on the stress concentration, in fatigue, due to stress intensifiers such as fillets, V-notches, square shoulders, small holes, and surface finish of specimens, and will indicate that for the materials tested the question of ductility, chemical composition, and heat treatment also play an important part.

Since wrought ferrous metals are used almost entirely for the stress-carrying members of machines and structures, these remarks will be confined to this general class of materials. It should be pointed out that for steel, the fatigue or endurance limits, by which stress intensifications are analyzed, are found almost without exception within the static elastic range of the materials considered. It should also be pointed out that the metals used have been in the main very homogeneous and produced consistent results. They had been selected with care for fundamental fatigue tests, had been carefully analyzed as to chemical composition, and considerable care had been taken with their heat treatments.

Early in the history of the Fatigue of Metals Investigation[2] (1919) the questions of the effect of fillets, V-notches, re-entrant angles, and surface finish of specimens were studied in conjunction with their effects on the fatigue strength of steel.[3] In the preliminary tests two steels were used, one a 0.49-carbon heat treated to the sorbitic condition and the other a 0.02-carbon steel which was annealed. Fig. 30 shows the shapes of the specimens used. For the sorbitic steel the variation of the endurance limits as between the specimens with a radius of one inch and the specimens with a ten-inch radius was shown to be practically negligible. The specimens with $\frac{1}{4}$-in. radius reduced the endurance limit about 8 per cent, the square shoulders reduced it about 51 per cent, and the V-notch about 60 per cent. The low-carbon annealed steel showed a similar effect with the exception that the variation in each case showed respectively a slightly lower percentage than

[1] Engineer of Tests, Fatigue of Metals Investigation, University of Illinois, Urbana, Ill.

[2] The Fatigue of Metals Investigation refers to the investigation on the fatigue of metals conducted by the Engineering Experiment Station of the University of Illinois in coöperation with the National Research Council, Engineering Foundation, the General Electric Company, the Copper and Brass Research Association, the Western Electric Company, and the Allis-Chalmers Manufacturing Company.

[3] See Reference 1 in the list on page 228.

was the case with the higher carbon heat-treated steel. It was considered that the square shoulder and the V-notch in the specimens, which according to the theory of elasticity added very much to the intensifying effect of stress, should have affected the endurance limits very much more than they did.

Another study was undertaken dealing with the effect of surface finish of specimens on the endurance limits [1] of steels. The same two steels were used in this study as were used in the previous one and the types of surface finish are indicated in Fig. 31.

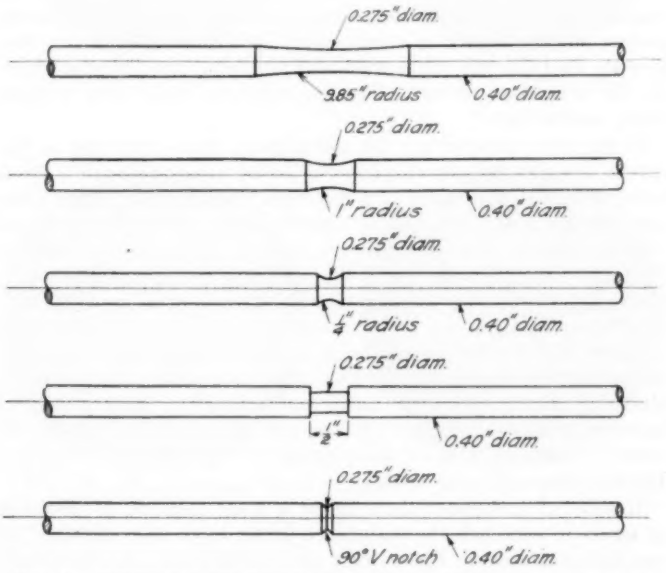


FIG. 30 SPECIMENS FOR STUDY OF EFFECT OF SHAPE ON ENDURANCE LIMIT

The type of finish finally adopted as the standard is the one marked (*d*). The rough-turned finish (*a*) had an endurance limit for the 0.49-carbon steel about 18 per cent below the standard finish. The effect of the intermediate finishes was found to fall between the results of the rough-turned (*a*) and the standard finish (*d*). The effect of the rouge finish (*e*) showed practically the same results as the standard finish and since the rouge finish consumed very much time in the preparation of specimens the finish marked (*d*) was adopted as the standard for use in the investigation. The tests on the effect of surface finish of specimens were continued on the low-carbon annealed steel with the

[1] See Reference 1 in the list on page 228.

result that the effect was found to vary slightly less in each case than for the steel with the higher percentage of carbon. It was considered, therefore, that ductility and heat treatment were factors which mainly influenced the difference in results obtained from the two steels tested.

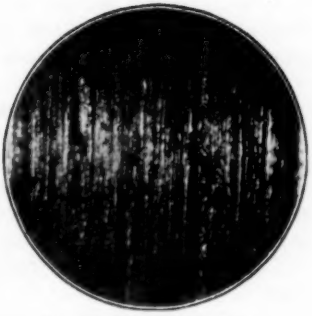
Later, when the nature of the endurance limits for steel became known, and when it was understood that the endurance limits were obtained from specimens which did not fail — specimens which were not overstressed — and that the endurance limit stress occurred in almost every case within the static elastic range of the metal, the idea of using discontinuities in the shape of specimens to test out the theory of elasticity as applied to fatigue was conceived. In 1923 this conception was put into experimental effect. In the meantime certain other investigators have been working along similar lines.[1]

It has been pointed out by the authors that, according to the elastic theory, the concentration of stress at the edge of a small circular hole in the center of a flat tension specimen is three times the average stress if the same load is used on a similar specimen without a hole. Reference to work done on the elastic problem by Professor Coker [2] in the use of the photoelastic method shows that statically, within the elastic range of the material used by him, this theory is borne out very closely indeed. The work done using Lueders' lines on steel, which indicate the yielding of the materials, shows that for ductile steels within the elastic range, the theory of elasticity is not so closely borne out statically. This condition, however, may be due partly to the fact that there is considerable relief of stress because of the yielding of the material before the Lueders' lines show up readily.

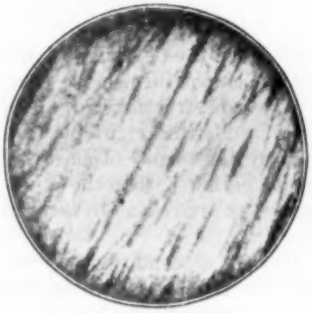
Results of endurance tests using a large number of reversals of stress to establish the endurance limits have been obtained on ten different steels. These tests have established the endurance limits on plain specimens, the apparent endurance limits on specimens with a small hole through the center, and the apparent endurance limits on specimens with $\frac{1}{8}$-in. fillets. The apparent endurance limits have been calculated without considering the intensification of stress due to a small hole or a fillet. The assumptions made in those tests are: that the actual endurance-limit stresses present in the specimens with holes or with fillets were equal to the endurance-limit stresses obtained in the plain specimens; that as soon as the stress at any point in the specimen is above the endurance limit of the material it is simply a matter of time before the specimen fails; and that if failure does not occur until 10,000,000 cycles of stress have been applied it will not occur at all. Fatigue results over a period of five years have shown

[1] See References 5 and 7 on page 229.
[2] See Reference 6.

(*a*) Rough-Turned Finish

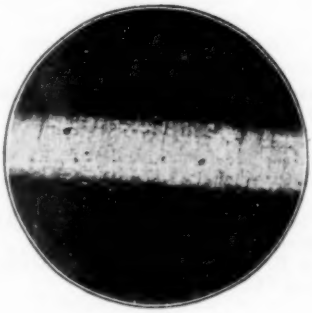


(*b*) Smooth-Turned Finish



(*c*) Ground Finish

(*d*) Standard Finish



(*e*) Rouge Finish

FIG. 31 MICROGRAPHS OF SURFACE FINISH

Magnification 30 Diameters

that for wrought ferrous metals at atmospheric temperatures, these assumptions are sound, providing that a commercially homogeneous steel is available from which to make specimens for a set of tests.

The apparent endurance-limit values obtained on specimens with a small hole were divided by the endurance-limit values on plain specimens to show the percentage effect of holes in intensifying a stress at the edge of holes. According to the theory of elasticity, the value obtained in this manner should be 33.3 per cent. The ratio for the ten steels tested, however, varied from 56.7 per cent to 81.8 per cent, with an average value of 75.1 per cent.

The apparent endurance-limit values obtained on specimens with fillets of $\frac{1}{8}$-in. radius divided by the endurance limit on plain specimens varies from 67.7 per cent to 97.0 per cent, with an average value of 82.6 per cent. Statically within the elastic range of the material, Professor Coker has shown that this ratio should be 69.6 per cent.[1]

It is apparent, therefore, that, within the elastic range of the material tested, the theory of elasticity does not furnish the engineer with accurate data on which to design structures in which repeated stress at points of high localized stress is to play an important part. The values computed by the theory of elasticity are on the safe side for steel. It should be noted that the above observations are in accord with conclusions arrived at by several other investigators.

It is also apparent that for any fixed set of conditions the intensification of stress, using endurance limits as an indicator, will vary with the composition of the material and with the heat treatment. It is suspected that the relative size and shape of the piece tested will have a material effect on intensification of stress in fatigue tests even though the maximum stresses involved are well within the static elastic range of the tested material.

List of References

1 H. F. Moore and J. B. Kommers, Bulletin 124, Investigation of the Fatigue of Metals, Engineering Experiment Station, University of Illinois, 1921.

2 H. F. Moore and T. M. Jasper, Bulletin 142, Investigation of the Fatigue of Metals, Engineering Experiment Station, University of Illinois, 1924.

3 H. F. Moore. Fatigue Tests of Metals and the Theory of Elasticity, *Engineering News-Record*, Feb. 5, 1925.

4 J. S. Wilson and B. P. Haigh, Stresses in Bridges, British Association Stress Committee, 1923.

[1] The experimental values bearing out the preceding summary were published in Bulletin 152 of the Engineering Experiment Station of the University of Illinois, 1925.

5 B. P. Haigh and A. Beale, The Influence of a Circular Hole on the Fatigue Strength of Hard Steel Plates, British Association Stress Committee, 1924.

6 E. G. Coker, Photoelastic Methods of Testing, British Association Stress Committee, 1924.

7 R. R. Moore, Resistance of Metals to Repeated Static and Impact Stresses, A.S.T.M. Proceedings, 1924.

8 W. N. Thomas, The Effect of Scratches and of Various Workshop Finishes upon the Fatigue Strength of Steel. Memo. No. 860, Aeronautical Research Committee (British).

9 A. A. Griffith, Stress Concentration in Theory and Practice, Complex Stress Committee, British Association, 1921.

10 H. J. Gough and D. Hanson, Behavior of Metals Subjected to Repeated Stresses. Proceedings, Royal Society, 1923.

11 C. E. Inglis, Stresses in a Plate due to Presence of Cracks and Sharp Corners. Transactions, Institution of Naval Architects, 1913.

HENRY S. PRICHARD.[1] The writer does not take part in this discussion with entire freedom as some matter which is new and which he considers would be pertinent to this discussion is embodied in a paper which has been accepted for publication but not yet published by another engineering society.

It is well that engineers should be reminded occasionally by papers like the authors' that the effect on a structural or machine part, when loaded, of holes, sharp reëntrant corners, and sudden reduction in cross-section, is not simply to increase the intensity of the stress in the same proportion that the cross-sectional area is reduced, but also to change its distribution and produce stress concentrations which, with some kinds of material and in some circumstances, may be very serious. When so reminded it is natural that engineers should want to know what to do about it.

The authors conclude, for ductile materials, that the stress concentration from holes, grooves, and sharp variation of cross-section does not have a weakening effect under static loading but that under the action of stress reversals its weakening effect may become of prime importance and must be taken into consideration. They further conclude that "the weakening effect of stress concentration in the case of ductile material submitted to the action of stress reversal can be established accurately enough only on the basis of fatigue tests of the models of the machine parts under consideration."

The load in very few cases is entirely static. In most cases there is a live load which causes the combined stress to fluctuate between limits which may both be in tension or both in compression or one in tension and one in compression. Under combined live and dead loads, the utmost which a structural or machine part which is made of ductile material and which is not lacking in rigidity, will in long service satisfactorily endure, depends on the elastic limits developed during, and the distribution of stress produced

[1] American Bridge Company, Pittsburgh, Pa.

by the load on and after, the recovery of elasticity subsequent to plastic deformation within permissible limits. When the stress is not reversed, stress concentration may be lessened or even eliminated by such plastic deformation; and even if reversed but not completely reversed it may be somewhat lessened. The elastic range finally developed will be greatest under complete reversal of stress and will be least, other things being equal, in the case which has the greatest maximum stress. The most reliable criteria are endurance or, as they are termed, fatigue tests. For the same kind and composition of material, rolled or forged bars or shapes which, as rolled or formed, have smaller cross-section than other rolled or forged bars, will generally have greater elastic strength. This fact should be considered when gaging strength from tests of models of structural or machine parts made of smaller rolled or forged bars of the same material.

The authors reach the conclusion that " in the case of brittle materials the weakening effect of stress concentrations is given with a good approximation by the factor of stress concentration which can be established analytically or experimentally by one of the methods that have been described." Even cast iron and very hard steel, which are usually classed as brittle materials, have some ductility which tends in some cases to reduce stress concentrations. The writer would prefer endurance tests and experience to mathematical analysis, but where there is a deficiency in these regards the methods and tables which the authors present should prove useful. There is one point, however, to which attention is invited: The authors seem to regard resistance to tension and compression as the criteria of strength in cases of compound stress. The writer's studies have led him to consider that elastic failure under tension or compression is really elastic failure by sliding or shearing along certain planes under the shearing component in the direction of those planes of the tension or compression. In cases of simple tension or compression it makes no difference whether the simple stress, be it tension or compression, or the maximum shearing component of the simple stress is considered critical, but for cases of combined tension or compression in two perpendicular directions, if the limiting condition is inferred from experiments in simple stress, it does make a difference as to what is actually the limiting condition. For instance, in the case of a rectangular body with a longitudinal tension and transverse compression of equal intensity as compared with the same body under the tension alone, the intensity of the tension will be the same, the longitudinal strain, taking Poisson's ratio as 0.3, will be 1.3 as great and the intensity of the shear will be twice as great. To resist the compression in addition to the tension, the body would, the writer considers, require twice the section. The writer, however, has not attempted to solve with shear as the criterion the problems of compound stress presented in the paper.

Photoelasticity applied to the fillet problem and Lueders' line method applied to problems of fillets and holes, as handled by the authors, are quite interesting. The use of celluloid for stress determinations by photoelasticity is in its infancy and gives promise of becoming very useful.

GEORGE F. SWAIN.[1] This is an interesting paper dealing with an important matter, but one in regard to which there is much misapprehension. The authors state that it is possible to find the stress by an analytical method "for such *simple* cases as circular or elliptical holes," and they give an approximate method for doing this. (Italics the writer's.)

In the case of a circular hole they state that "the stress distribution near the hole may be obtained from the general equations of the theory of elasticity," and they refer (Par. 12) to what they term "the exact solution," which gives the stress intensity at the edge of the hole as three times the average intensity on the gross section for the case in which the diameter of the hole is d and the width of the plate infinite. They do not give the formulas on which this is based, but presumably they are those given in a paper by Coker and Scoble in the Transactions of the Institution of Naval Architects for 1911, quoted from Föppl's Vorlesungen über Technische Mechanik, vol. 5, p. 352. This is the only claimed solution of this case that the writer has ever seen. It may be worth while to examine first this so-called exact solution. It is based on two assumptions. The first, of course, is that there is a hole. The second is that the width of the plate is infinite. The last assumption means that there is no hole at all, because a hole of finite diameter d in a plate of infinite width is the same as a hole of no diameter in a plate of finite width w, since

$$\frac{d}{\text{infinity}} = \frac{0}{w}$$

Here we have an instance of the character of some of the demonstrations that are now being put forward as founded on the theory of elasticity and as being "exact." This one, as above stated, is founded on the assumptions that there is a hole and that there is no hole; in other words, that a thing is and is not at the same time. It gives results for the stress at the edge of the hole which are independent of the diameter of the hole. Of course, it is easy for any practical man to see that such results are absolutely worthless as applied to any practical case.

But we may proceed to examine the "approximate solution" which the authors give. They treat the case as one of a circular ring, the inside diameter of which is the diameter of the hole and the outside diameter the width of the plate. They imagine a circle

[1] Professor of Civil Engineering, Consulting Engineer, Harvard University, Cambridge, Mass. Mem. A.S.M.E.

drawn around the hole having this outside diameter, and treat the case as a circular arch or ring. Of course, in treating such a case it is necessary to know the outer forces applied to the ring around its circumference, and these outer forces the authors assume to be distributed uniformly across the width of the plate, as shown by Equation [1]

$$p = p_0 \sin \phi$$

In other words, on any length ds of the circumference they assume the stress to be longitudinal and equal to $p_0 ds \sin \phi$, which is the projection of ds on the transverse section multiplied by the uniform intensity p_0 in the end section of the plate.

The first criticism to be made of the paper itself is that this fundamental equation, on which all the rest of this demonstration is based, is incorrect. The longitudinal stress would not be uniformly distributed transversely around the outer semicircle of the ring. We do not know how it is distributed. Nobody does.

On the basis of this incorrect assumption, the authors proceed to compute the distribution of the stress on the transverse section through the center of the hole or ring.

There is a second error, however, in their treatment. In Fig. 2 they show a quadrant of the ring with the forces which they assume are acting upon it. As above stated, they assume the stress on the outer circumference to act longitudinally in the direction of the pull in the plate, and they assume no normal or horizontal (transverse) stress on the upper section of the ring. Their so-called "exact" theory gives a normal stress on this section, and consequently horizontal stresses on the outer semicircumference. They refer to this normal stress in Par. 14 and Fig. 4, where they state that at the points a there is a transverse stress of compression equal to the tensile stress p_0 at the ends of the plate, but neglect it in Fig. 2.

In the writer's book on Strength of Materials he discussed this matter also and showed that on a longitudinal section through the center of the hole there must be normal stresses. These stresses will be compression at or near the edges of the hole and tension at points farther removed, the tensile and compressive stresses between the hole and one end of the plate forming a couple.

In Article III the authors consider combined tension or compression in two perpendicular directions, using Lamé's formula for thick cylinders. This formula is frequently used, and perhaps there is no other that is better. It may be pointed out, however, that it involves an assumption which is not necessarily true. This part of the paper is ingenious but is also founded on error.

Regarding these investigations in the theory of elasticity, it may be well to quote from the distinguished mathematician and historian of the theory of elasticity, Karl Pearson, editor of Tod-

hunter's History of the Theory of Elasticity and of the Strength of Materials, who said (vol. I, page 626):

> We find even in mathematicians of the standing of Lamé not infrequently an omission to state clearly the physical principle upon which they base their calculations of a physical phenomenon. The history of mathematical elasticity gives many examples of this divorce between theory and physical fact; the mathematician has too often identified elasticity with the solution of certain differential equations, the constants of which are to be determined by a purely fanciful and often practically idle, if not impossible, distribution of load.

The treatment of a circular hole with a bead involves the same errors above specified. The case of an elliptical hole is briefly mentioned as though this case had been solved by the formula given, for the stress at the edge of the hole gives the same result for a circular hole as the so-called exact formula which the writer has criticized; so that it probably involves the same assumptions.

The authors proceed to discuss photoelasticity as applied to this problem. This is a method of investigation which has recently achieved some prominence because it is interesting and involves some pretty color effects. How much reliance is to be placed on the results, however, is, in the opinion of the writer, entirely uncertain. The writer discussed this subject also in his Strength of Materials, and stated that he was not prepared to accept the results, giving his reasons. The following sentences are from his book:

> But since the color depends only on the difference of phase of the vibrations parallel to the principal axes, and not upon the actual retardation, there must be an indefinite number of values of the difference of principal stresses which would give the same color effect. The writer, therefore, is unable to see that these experiments can give quantitative results, though they may give relative results in some cases. Professor Coker himself says, "It is quite possible for a material to be highly stressed at a place, and yet exhibit little or no color effect," this referring, no doubt, to the fact that with equal principal stresses there would be no color. He also says, "There is no certain way of deciding whether the part under inspection is in tension or compression."

The writer also stated that while the field was a promising one for research, he was not yet prepared to accept the results. In this connection it is interesting to observe that in a review of his book in an English magazine[1] the reviewer said: "We note with considerable interest *and agreement* that Professor Swain states that he cannot accept the results made by testing on the photoelasticity method." (Italics the writer's.) It seems, therefore, that English structural engineers are not prepared to accept the results, and they should not, therefore, be put forward as definite.

The authors, in Article VIII, refer to other measurements of strain from which stress is inferred, notably those by Professor Preuss. They tell us some things about Professor Preuss's tests,

[1] *The Structural Engineer*, March, 1925, published by The Institution of Structural Engineers of England.

some of which they reproduce. They tell us, for instance, that the tensile force in all cases was such that the average stress on the weakest cross-section was equal to 750 kg. per sq. cm. This is true, but of no consequence. It is of no importance whether the average stress was 750 kg. or more or less as long as it was within the elastic limit. They omit, however, some facts with reference to Professor Preuss's tests which are of real importance; namely, the fact that in these tests the gage length for his extensometer was as small as 7/10 of a millimeter, or less than 3/100 of an inch. In some cases it was 3 3/10 millimeters. We wonder how many engineers who look at the curves from these tests, and perhaps give some credence to them, know that they were made on such minute gage lengths, and how many would be disposed to rely on these results if they knew that fact. It is easy to fasten an extensometer on a test specimen and observe the reflection from a mirror. It is quite another thing to judge to what extent the results mean anything, and what they mean. The best results of tests in this matter which are known to the writer are those of Professor Rudeloff, published in 1915. He tested a plate with two holes in it. His shortest gage length was 0.59 in. and his results did not agree with those of Preuss.

In the judgment of the present writer, engineering today is being and has been demoralized by the abuse of mathematics and of testing. Mathematics is an invaluable tool, a necessary tool, but it is a dangerous tool, because the tool itself is so interesting that those who are expert in its use but do not understand the meaning or the physical limitations of the problems to which it is applied will misuse the tool.

It is the same with testing. There is, of course, no way by which we can discover the secrets of nature except by observation and tests and by reasoning based upon them; but testing may be abused and misused. Many tests are made to discover things which can be reasoned out from known facts, and many other tests are made in which the inevitable errors are such as to render the results of no value, or in which the conditions are such that no definite conclusions can be drawn. The writer has elaborated these ideas in the last chapter of his book on the Strength of Materials, which is entitled Mathematics, Testing, and Common Sense, and this chapter may be of interest to those who take the trouble to read this discussion. It is a plea, which is here repeated with emphasis, and is believed justified by the paper under discussion, for the use of practical common sense instead of mathematics based on false assumptions, and testing based on incorrect conditions.

S. TIMOSHENKO.[1] The results obtained by Professors Moore and Jasper are in agreement with the results which we obtained in our laboratory.

[1] Authors' closure

In Mr. Prichard's discussion the fundamental question is raised as to the strength theory which must be taken as a basis in figuring working stress in the cases of combined stresses. In the case of structural steel, we always use the maximum-shear theory and think that it is in accordance with all experimental data on ductile ferrous materials. In our photoelastic experiments we measured stresses at the edge of fillets where one of two principal stresses becomes zero and there is only simple tension in the direction of the tangent to the edge of the fillet.

It is rather difficult to answer the discussion presented by Professor Swain. As the writer understands it, Professor Swain does not believe in the mathematical theory of elasticity, nor does he believe in the photoelastic method or the direct strain measurement by Preuss's extensometer. The exact solution of the problem on stress concentration produced by a circular hole in a plate of infinite width was obtained on the basis of the general equations for a two-dimensional problem of elasticity. From these equations it follows that a small hole in the plate of infinite width is not the same as no hole at all, as Professor Swain erroneously states in his discussion. On basis of these equations, it can be shown also that in the case of a plate having a large width, the stress concentration is independent of the diameter of the hole. An analysis made by the writer[1] shows that if the width of the strip is more than $5d$, the stress concentration is practically the same as for a plate of an infinite width.

Professor Swain thinks that "it is easy for any practical man to see that such results are absolutely worthless as applied to any practical case," but we know that these results are in very good agreement with the results of photoelastic investigations and with the results of direct measurements of deformation. Many practical engineers realize the importance of the phenomena of stress concentration and take it into consideration in their design work, especially in designing machine parts that will be subjected to the action of reversal of stresses.

Discussing the authors' approximate solution, Professor Swain finds Mr. Dietz's and the writer's assumption on stress distribution along the outer boundary of the ring (see Fig. 2) to be incorrect. But this assumption follows at once from the well-known principle of Saint-Venant on the basis of which they are justified in assuming that a circular hole produces a disturbance in the stress distribution of localized type; i.e., the effect of the hole on the stress distribution is essential only near the hole, and by taking a circle of large radius, as assumed in the paper, the effect of the hole on the stresses can be neglected. There is another justification for their assumption. For the case of a circular hole, an exact solution is available, and it shows that additional stresses produced by

[1] See Bulletin of the Polytechnical Institute in Kiev, 1907.

boring a hole diminish as the square of the ratio r/R, in which r = radius of the hole and R = radial distance of the point under consideration. If $R = 5r$, the effect of the hole on stresses is not larger than 4 per cent. This demonstrates that their assumption is correct.

The statement of Professor Swain that the presence of horizontal stresses on the outer semicircumference follows from the fact that there are normal stresses on the upper section of the ring, is incorrect. These stresses are produced by bending of the ring, and for this it is not necessary to have the horizontal stresses mentioned above. In his discussion Professor Swain makes a reference to his book on the Strength of Materials in which the problem on stress concentration is discussed in an elementary way; but by using a simple beam formula (see page 123, Eq. 13, of Professor Swain's book), it is impossible to disprove the exact solution. The errors in his reasoning have been indicated also by another author,[1] and the writer hopes that in the next edition of his book Professor Swain will give a more satisfactory discussion of such an important question as stress concentration produced by notches and holes.

Professor Swain thinks that the photoelastic method, "which has recently achieved some prominence because it is interesting and involves some pretty color effects," is entirely uncertain. The writer completely disagrees with this statement. He was working with this method as early as the beginning of the year 1906, and had sufficient evidence to come to the conclusion that the method can give not only "pretty color effects," but useful quantitative data on stress distribution. The color gives directly the difference of principal stresses, i.e., the maximum shear which is the most important factor in design. In the case under consideration, the stresses at the edge of the fillets were studied and here the color gives directly the maximum stress because the other principal stress is here equal to zero. The work was done with a celluloid model $\frac{1}{8}$ in. thick, in which only colors of the first order were present, and there was no question at all on colors of higher order. It is easy to see also that by using a comparison member there is no difficulty in distinguishing between tension and compression.

The criticism of Professor Swain relating to Preuss's extensometer seems to the writer to be unjustifiable. The writer has had some experience with this instrument. In Preuss's work in measuring stress distribution which Professor Swain mentions, a 3.3-mm. gage length was used. The writer used an instrument with 10-mm. gage length in measuring stresses in the web of I-beams and found it as reliable as the well-known Martens extensometer. If the same reading is obtained after loading and unloading several times, it would seem that there are no displacements in the instrument other than those due to the deformation being studied.

[1] See *Engineering*, July 31, 1925, page 144.

It is believed that all the questions raised by Professor Swain have been answered. In conclusion, the writer desires to make some remarks in general about analytical and experimental methods in modern technical literature. The trend of modern industrial development is more and more toward the free acceptance and application of the teaching of pure science. This general tendency can be seen also in the increased use of the mathematical theory of elasticity for solving technical problems. In many cases of modern design the elementary solutions obtained by the application of the theory of strength of materials are insufficient, and recourse has to be made to the general equations of the theory of elasticity in order to obtain satisfactory results. All problems on stress concentration are of this kind. They involve highly localized stresses and elementary methods like those given in Professor Swain's book, pp. 122 and 123, cannot give a satisfactory solution. Only a complete analysis of stress distribution, together with experiments such as discussed in the authors' paper, can be expected to yield sufficient data for a practical design.

No. 1959

LAKE WATERS FOR CONDENSERS

By A. G. Christie,[1] Baltimore, Maryland
Member of the Society

The paper points out the persistence of stratification of warm- and cold-water layers in our northern lakes and other deep waters. The cold bottom layers make excellent cooling-water supplies for condensers and would permit operation at high vacuum throughout the whole year. The savings from increased vacuum are such that engineers should give this factor careful consideration, both in the selection of plant site and in the design of intakes, particularly for large power stations. Temperature-depth studies of the particular lake should be made to furnish exact data upon which a rational decision can be based.

ENGINEERS have accepted as a matter of common experience the fact that water taken from the lower depths of lakes is colder than surface water, particularly during the summer season. Intakes for cooling water to condensers have been located well below minimum low water to take advantage of such colder water as may be available. When large power stations are located on lakes or other large bodies of quiescent water in the northern section of this country, there are certain phenomena connected with the stratification of such waters that should be given careful consideration in the design of cooling-water intakes. This paper will discuss these phenomena and will indicate the large economic savings that may be realized by the proper design of intakes to take full advantage of the cold waters that are found in the lower strata of these lakes.

2 A large number of data concerning the temperature of our northern lakes have been collected and published by biologists in their study of fish life and of other organisms and by water-supply

[1] Professor of Mechanical Engineering, Johns Hopkins University.

Contributed by the Power Division and presented at the Spring Meeting, Milwaukee, Wis., May 18 to 21 1925, of The American Society of Mechanical Engineers.

engineers. Most of the data in this paper concerning lake conditions have been drawn from these sources.[1]

Factors Affecting Temperature Variation in Lakes

3 The northern lakes are generally frozen over completely or have surface waters which drop to nearly 32 deg. fahr. for many months each winter. The surface if frozen will be at 32 deg. fahr. The water at the bottom of the lake is generally 34 to 37 deg. fahr. during the winter; and little mixing takes place since the water at this temperature is more dense than at 32 deg. fahr.

4 When spring comes the ice gradually disappears. The wind, being an active agent in breaking up the ice, drives it towards the shores and exposes the surface of the water to the sun's rays. Most of the radiant energy of the sun's rays is absorbed by the upper five feet of water if the surface is undisturbed. Hence the surface waters warm up quickly.

5 Lake waters are usually disturbed more or less in spring by winds, and these are the active agencies at certain times in mixing waters that may be at slightly different temperatures. However, the effects of the wind as a mixing agent are opposed by the thermal resistance to mixture due to the differences in density of the various strata of water in the lake. This effect becomes more noticeable later in the summer.

6 After the ice has left the lake, the surface waters warm up gradually to 39.2 deg. fahr., the lower water remaining practically unchanged in temperature. Water has its greatest density at 39.2 deg. fahr., and if the bottom water is at a lower temperature, say, at 35 deg. fahr., it will rise to the surface to be displaced by the dense surface water at 39.2 deg. fahr. This results in a complete displacement of the water at the bottom by surface water. Thus the water of these lakes is overturned twice a year, in spring and again in the fall. This overturn is brought about partly by the wind and partly by convection currents due to differences in density. The wind is an important factor in bringing about this overturn and particularly in continuing it. The dense surface water is often blown to the leeward side of the lake where it sinks to the bottom as a means of returning to maintain the water level on the windward side. This displaces the bottom water and starts currents which rapidly overturn the whole lake. The mixing process continues until practically all of the water is mixed up and the whole of the lake approaches a constant temperature.

[1] An Unregarded Factor in Lake Temperatures, by E. A. Birge, Wisconsin Academy of Sciences, 1910. The Inland Lakes of Wisconsin, by E. A. Birge and C. Juday, Bulletin XXII, Scientific Series No. 7, Wisconsin Geological and Natural History Survey, 1911. Microscopy of Drinking Waters, Geo. M. Whipple; J. Wiley and Sons. The Temperatures of Lakes, Desmond Fitzgerald, Trans. A.S.C.E., 1895, and others.

7 The circulation often continues for a month or more in many lakes, for although the surface waters heat up in day time, they cool off at night and the convection currents are started again. This circulation frequently continues until the whole lake has increased to 45 deg. fahr. in temperature. However, at some time in the late spring the sun will warm the surface waters so that their temperature will not fall during the night below that of the lower layers and as soon as this happens circulation ceases. The lower waters never rise again to the surface of the lake during the whole summer, and, as will be shown later, their temperature throughout that period rises only a few degrees above that at which circulation stopped. This is a very important fact to power engineers.

8 The value of the thermal resistance to mixture is of interest for it shows why the winds can no longer cause mixing of the waters at various depths. The thermal resistance depends solely on the change of density of the water as its temperature rises above 39.2 deg. fahr. At first the change in density for each degree rise is small but this change per degree increases rapidly with rise in temperature above 39.2 deg. fahr. The following example will serve to show the increasing rate of change of density: A given volume of water weighing 1 lb. at 49 deg. fahr. weighs 0.000045 lb. less at 50 deg. fahr. for the same volume; another volume weighing 1 lb. at 76 deg. fahr. weighs 0.000142 lb. less at 77 deg. fahr. for the given volume. The rate of change in density per degree at 77 deg. fahr. is thus over three times the rate of change at 50 deg. fahr., and so thermal resistance per degree is increased.

9 Studies of density changes indicate that the thermal resistance to mixing is small for a few degrees either side of 39.2 deg. fahr., so that the lake waters frequently reach temperatures of 40 to 45 deg. fahr. before the spring overturn ceases, particularly if strong winds prevail. However, increasing surface temperatures soon increase the rate of change of density so rapidly that this mixing must cease. The wind effects do not penetrate to the lower depths. Convection currents due to differences of density have ceased and water is a poor conductor of heat. Hence the layers near lake bottoms receive no further heat and remain undisturbed at practically the same temperature throughout the whole summer and well into the fall. The temperature of the bottom layers throughout the summer depends on the occurrence of a period of hot weather early or late in the spring. The later the hot weather occurs to rapidly warm the surface layers and to stop convection currents, the higher will be the temperature of the lower layers when circulation ceases.

10 Observations taken upon lakes throughout the summer have indicated the same temperature as at the surface for the first few feet and often for 10 or 12 feet in depth. This would indicate that the winds keep the water stirred up more or less throughout

this depth. Below 12 feet the temperature decreases until at a lower depth depending on the particular lake, the cold strata are reached where the temperature is constant for the whole summer. The coldest waters of the lake are encountered at different depths depending on the deepness of the lake. This depth varies from 25 to 30 ft. from the surface for lakes about 65 feet deep while at Lake Thun in Switzerland with a total depth of 550 ft. the cold layers start at about 125 ft.

11 The discontinuity layers of varying temperature that lie below the warm surface water, have been called the " thermocline."

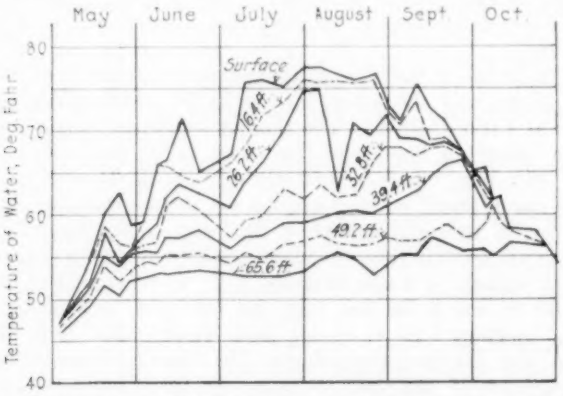


FIG. 1 TEMPERATURE VARIATIONS WITH DEPTH IN LAKE MENDOTA

The fall in temperature at the junction of the thermocline and the warm surface layers is often very rapid. Observations have fre-

TABLE I TEMPERATURES OF GREEN LAKE, WIS., IN DEGREES FAHRENHEIT

Depth in Feet	Temperatures Deg. Fahr.								
	Aug. 20, 1905	Feb. 15, 1906	May 15, 1906	July 5, 1906	Aug. 14, 1906	Oct. 4, 1906	Sept. 14, 1907	Sept. 3, 1908	Aug. 20, 1909
0.0	73.4	32.0	...	67.6	76.1	64.2	65.7	68.7	78.1
32.8	69.2	33.5	...	66.2	68.2	63.9	63.5	68.3	70.7
39.4	63.1	...	...	56.8	61.7	63.9	57.4	64.4	60.1
49.2	54.7	...	...	50.2	54.1	54.1	49.5	52.3	53.2
65.6	47.5	33.5	42.1	45.1	46.8	46.6	45.8	45.8	46.4
98.4	45.3	34.0	42.1	43.9	44.2	44.0	43.9	44.2	44.8
131.2	43.1	34.7	41.2	43.3	44.2	43.2	42.4	42.6	44.4
152.4	42.8	35.0	40.6	42.8	43.	42.6	42.1	42.4	44.1
182.9	42.4	35.4	40.5	42.3	42.4	...	41.7	42.3	43.7
213.4	42.3	36.1	40.5	41.5	42.3	42.3	41.4	41.9	43.7

quently indicated a drop at this junction of 7 to 9 deg. fahr. in three feet of depth when the surface temperatures are above 60 deg. fahr. In some cases there even appears to be a sharp dividing line between the surface water and the thermocline. Such a drop in temperature offers a very great thermal resistance to mixture by convection. There is a much less abrupt change in

temperature between the layers of cold water at the bottom and those immediately above them as is indicated by accompanying curves and tables. The position of the thermocline when once established is altered little throughout the summer, even by violent and long continued winds.

12 Mercury thermometers are not well suited for the measurement of water temperatures at different depths. Sensitive electric resistance thermometers or thermocouples have proven best for this service.

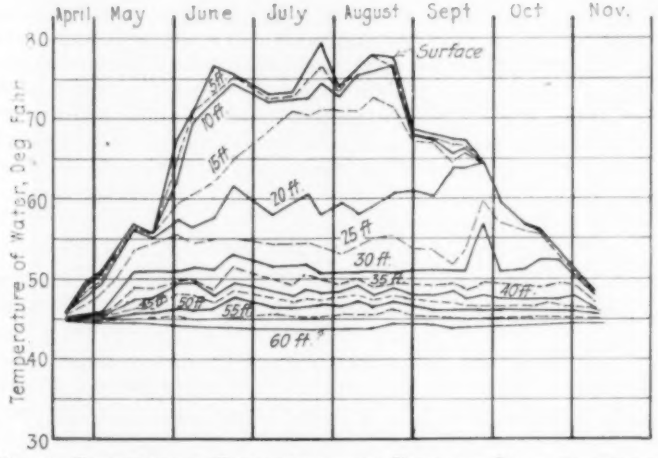


FIG. 2 TEMPERATURE VARIATIONS WITH DEPTH IN LAKE COCHITUATE

TEMPERATURE-DEPTH DATA

13 Fig. 1, reproduced from The Inland Lakes of Wisconsin, shows the variation of the temperature of the water at various depths in Lake Mendota. This lake has an area of about 15 square miles of which about 4.5 square miles is 60 ft. or more in depth. Apparently the vernal overturn ceased at 45 deg. fahr.

14 Table 1, also from The Inland Lakes of Wisconsin, shows the variation of temperature with depth in Green Lake. This is the deepest inland lake in Wisconsin. Its area is about 11 square miles. Sixty per cent of this area is 65 ft. or more in depth, while 50 per cent is over 100 ft. in depth. The lake bottom rises steeply within a few hundred feet of some of its shores so that its lower waters would be easily accessible for power-plant use. An interesting point indicated by this table is that the temperatures of the lower strata while constant throughout each season may vary slightly in different years, depending on the temperature at which the spring overturn ceased.

15 Fig. 2, from The Temperature of Lakes, by Desmond Fitzgerald, shows graphically the variation of temperature with depth for Lake Cochituate near Boston. The undisturbed, cold lower layers are apparent. This paper also contains some observations made on Lake Superior which show that with surface temperatures of 50 to 56 deg. fahr. the deep-water temperatures range from 36.8 to 39 deg. fahr.

16 Data on temperatures of many other lakes are available in Fitzgerald's paper, in The Inland Lakes of Wisconsin, and in various other publications. Certain lakes in Scotland and Switzerland have been studied and show the same temperature stratification. These all indicate the same phenomenon of undisturbed cold water near the bottom of all deep lakes during the summer months. Shallow lakes or other shallow bodies of water not exceeding 25 ft. in depth, are not likely to exhibit this phenomenon for they are too much stirred up by the winds. Temperature stratification will not likely occur in rivers for the currents mix these up too much although exceptions may be found in deep, slowly flowing rivers.

Temperature Variations in Bays, Sounds, and Ocean

17 Through the courtesy of Dr. R. P. Cowles of Johns Hopkins University, certain temperature-depth observations of the Hydrographic Survery, Bureau of Fisheries, on Chesapeake Bay were made available for study. Unlike the lakes just described, there does not seem to be any definite thermocline in the bay nor any cold bottom layers. There is some variation of temperature with depth but it is not large. This seems rather surprising until one studies the physical features of Chesapeake Bay. Though large in area and with deep channels, the average depth is not very great. It has strong currents from tides and receives large discharges from fresh-water rivers. The winds stir it up easily in summer and mix its waters. Thus every agency works against the establishment of a distinct stratification of its waters.

18 Observations made outside Cape Henry in the Atlantic Ocean indicated quite a different state of affairs. With surface temperatures of 70 to 75 deg. fahr. the temperatures at 60-ft. depth averaged about 42 deg. fahr. and this persisted quite close in to shore. One can therefore expect to find stratification along the sounds and bays of our coast where reasonable depths occur and where tides and currents do not disturb the waters too much.

Application to Condensers

19 The intakes for cooling water at power plants located on such bodies of water as above described, would probably lie from 15 to 20 ft. below the normal surface. Such intakes will draw their water from the thermocline where the temperature varies during the season, as shown by Figs. 1 and 2. A study of the

data here presented and of other available data indicates that water 10 to 20 deg. fahr. colder could be obtained by intakes that would reach the cold bottom layers. Power stations on such lakes usually carry vacua ranging from 28 to 28½ in. during the summer months. An inch of vacuum can be allowed for every 22-deg. fahr. change in intake temperature. With deep-water intakes and colder water it seems probable that 29 in. vacuum or higher could be maintained throughout the whole year. The possibility of such a condition is illustrated by Fig. 3 which is plotted from the observations in Table 1 for Feb. 15 and Aug. 14, 1906. The location

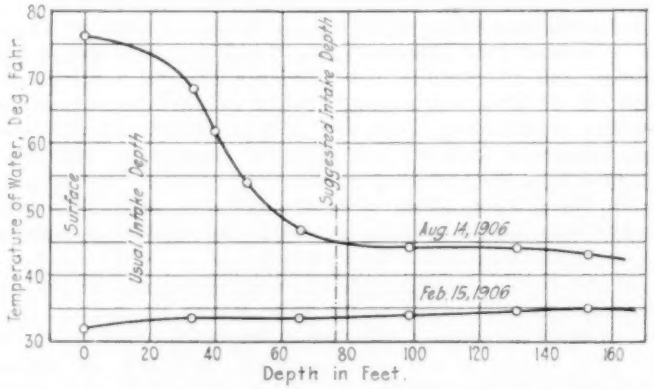


FIG. 3 RELATION BETWEEN TEMPERATURE AND DEPTH ON TYPICAL MID-WINTER AND MID-SUMMER DAYS IN GREEN LAKE, WIS.

of a condenser intake that would draw water from 75-ft. depth instead of 20-ft. depth would provide on the August day cooling water at about 45 deg. fahr. instead of 73 deg. fahr. This would develop an inch and a quarter more vacuum on the condenser. Furthermore with the deep intake there would only be a possible variation of cooling-water temperature from about 34 deg. fahr. in winter to 45 deg. fahr. in summer which would permit the design of a plant for at least a 29-in. vacuum throughout the year.

20 The decrease in steam consumption of a large turbine for a change in vacuum from 28 to 29 in. is about 7½ per cent, and from 28½ to 29 in., about 4 per cent. The value of additional vacuum due to colder cooling water may be indicated by the following figures assuming that the average vacuum was increased only from 28½ to 29 in. for 4½ months of the summer. Take a plant capacity of 100,000 kw.; a use factor over this period of 40 per cent; fuel consumption of 1½ lb. of coal per kw-hr.; coal cost, $5.00 per ton of 2000 lb.; and 4 per cent decrease in steam consumption due to increase of vacuum from 28½ to 29 in.

Savings for 4½ months

$$= \frac{4.5 \times 8760 \times 100{,}000 \times 0.40 \times 1.5 \times 0.04}{12 \times 2000}$$

= 3942 tons of coal valued at $19,710

21 If the vacuum had been increased from 28 to 29 in. with 7½ per cent less steam consumption the saving would have amounted to $36,956. No allowance has been made in these figures for the lower cost of pumping as a result of the probable use of smaller quantities of colder water at light loads.

22 The rate at which these savings of $19,710 and $36,956 per annum could be capitalized depends upon cost of money, taxes, and other conditions, among which are the number of years of service to be secured from such extra length and depth of tunnels and intakes, and maintenance and possible alterations in the meantime. If all these conditions are such that the foregoing savings could be capitalized at 12 per cent, then an approximate additional expenditure of $164,000 and $308,000, respectively, would be justified for a plant of 100,000 kw.

23 No difficulty would be experienced with ice troubles from such a deep intake in winter nor with trash troubles in summer. A simple screening device would suffice.

24 One will naturally say that these lower depths are not accessible to most plants as the shore waters are shallow. This is frequently the case and open cuts to deep water are objectionable for they silt up quickly. Under certain conditions the savings from the colder water may warrant the construction of a waterworks type of tunnel with a deep-water intake. If the distance is not great, a steel or concrete pipe might be laid on the lake bottom to the deep water. The additional pumping head will only be that due to friction in the additional length of conduit, and this may be offset by careful design of the deep-water intake so as to eliminate most of the losses that now occur in an open-ended intake. The discharge tunnel for the cooling water from the condensers should spread out near the surface in a wide shallow mouth with low velocity so as to distribute the hot water only to the surface layers of the lake and not force it into the colder lower layers.

25 There is little danger of using up the cold bottom layers during the season as will be shown by the following example. Assume 120 gal. of cooling water per kw-hr. and 10 sq. mi. or 6400 acres of cold water. The above station would develop 131,400,000 kw-hr. during these 4½ summer months, requiring 15,768,000,000 gal. or 2,100,000,000 cu. ft. A square mile contains 27,878,400 sq. ft. of surface. Hence the lower waters would be pumped out during the season for a depth of 7.5 ft. over the 10 sq. mi., or 15 ft. over 5 sq. mi.. Probably any lake would have over 5 sq. mi. of the colder waters available and hence there would be an ample supply of such cold condensing water.

26 The question may also arise whether only the colder water could be pumped out without also drawing in some of the upper and warmer layers. It is true that the colder water is somewhat more viscous than the upper layers. On the other hand, these lower layers persist in maintaining their stratification and their levels. With a proper design of intake it should be possible to take in only the colder layers.

27 During the summer period there is a tendency for the organic matter from the upper layers to collect in the lower layers, and this organic matter would pass into the condenser. Organic material gives little trouble from slimes and growth in condenser tubes with cold condensing water, and hence would be of no consequence on account of the low temperatures.

Conclusions

28 The persistence of stratification of warm- and cold-water layers in our northern lakes and other waters has been pointed out in the preceding paragraphs. The cold bottom layers make excellent cooling-water supplies for condensers and would permit operation at high vacuum throughout the whole year. The savings from increased vacuum are such that engineers should give this factor careful consideration, both in the selection of plant size and in the design of intakes, particularly for large power stations. Temperature-depth studies of the particular lake should be made to furnish exact data upon which a rational decision can be based.

DISCUSSION

C. G. Spencer.[1] The author, by developing the theory of the thermocline and applying it in a concrete example to central-station economics, has made a valuable contribution. Power engineers are interested in this subject probably to a greater extent than any other group of men, yet a recent search for published data disclosed none from this source.

Now that the importance of determining the thermocline has been pointed out, it is to be hoped that engineers having to do with condensing stations on our northern lakes will take temperature readings at varying depths during the six warm months of the year and publish the data.

The coal saving because of better vacuum in the summer months is attractive in plants of all sizes. To illustrate this further, assume a plant of 270,000 kw. capacity with a 50 per cent daily load factor.

[1] Engineer, McClellan & Junkersfeld, Inc., New York, N. Y. Mem. A.S.M.E.

A reduction of 20 deg. fahr. in condensing-water temperature means a daily coal saving as follows:

Cost of coal, dollars.....	4	5	6
Daily saving, dollars.....	500	600	700

The curves in Fig. 2 for a lake near Boston indicate that an intake 30 ft. down would give this 20-deg. lower temperature water during the three hottest months. The saving is not confined to coal only. By reducing the usual wide variation between winter and summer condensing-water temperature, specifications can be written to reduce the investment cost of condensers and condenser auxiliaries.

N. G. REINICKER.[1] The company with which the writer is connected has not had any experience along the lines discussed in the paper, but we believe that the data presented are good. One of our power plants depends upon an artificial lake for condensing-water supply. This has a maximum depth of 25 feet at the lower end. The condensing water is recirculated, being put back into the lake from a separate discharge canal, extending more than one-half mile upstream from the plant. We find that at least 90 per cent of the cooling effect takes place at the point where the water passes from the discharge canal in a thin sheet over some riprapping into the lake. We have made temperature surveys of the lake at various points, approximately one foot below the surface, and did not find a variation of more than 2 deg. fahr. at any point from the discharge canal to the lower end of the dam, where the water is again taken into the plant.

JAMES MILNE.[2] If full advantage is to be taken of the colder water available at certain depths, the intakes must be designed so as to give low entrance velocities. A high entrance velocity will cause a vortex which extends upward to form, and water many feet above the intake will be drawn in.

The City of Toronto has two intakes, located off Toronto Island, Lake Ontario, 72 in. in diameter, operating at from one to two feet per second, and in spite of their depth (43 and 79 ft., respectively) there is evidence that water is drawn in from very near the surface. In the summer months a change in direction of the wind from south or west to north or east, or vice versa, will cause a change in 24 hours of as much as 16 deg. fahr. in the temperature of the water from the 43-ft. intake, and 10 deg. in the 79-ft. intake.

[1] General Superintendent, Pennsylvania Power & Light Co., Allentown, Pa. Mem. A.S.M.E.

[2] Mechanical and Electrical Engineer, City of Toronto, Canada. Mem. A.S.M.E.

The following are the average temperatures (6 readings per day) for the past ten years of air and water at Toronto, Ontario:

	Deg. Fahr.	
	Air	Water
January (1914-1924)	21.76	34.54
February	21.57	33.77
March	31.71	33.99
April	44.09	35.80
May	56.37	39.40
June	66.98	41.63
July	72.59	43.73
August	68.82	48.43
September	61.28	48.85
October	49.61	43.39
November	36.65	39.83
December	26.00	37.97

The water is taken from the intakes already referred to. It next passes through the filtration plant, then through conduits and tunnel. The tunnel is over 5000 ft. long and 100 ft. below the surface of the bay.

A recording thermometer is used for taking the temperature, the bulb of which is placed at a depth of 45 ft. in the tunnel shaft. This recording thermometer is tested periodically by a thermometer "certified correct."

The spring and fall turnovers are just as certain as the seasons come and go. The top waters appear to carry all débris with them down to the bottom. During these periods we use revolving screens. We have no difficulty in obtaining on three DeLaval 1800-hp. steam turbines a vacuum of 29 in. during the entire year.

ROBERT W. ANGUS.[1] The data and general discussion presented in the first eighteen paragraphs of the paper are of value and interest. That the average temperature of water in lakes decreases with the depth is well known, and cities drawing water from the Great Lakes always go as deep as possible with their intakes for the purposes of getting colder and purer water and to avoid ice trouble.

Some few years ago the writer had occasion to make a careful study of the effect of depth and location on the temperature of the water in Lake Ontario near Toronto, Canada. In that part of the study conducted in September and early October, a comparison was made between the temperature at the present Toronto intakes, which are in 69 ft. of water, and a proposed site some miles to the east where the depth was 49 ft. In that case, water at a depth of 45 ft. never showed a greater temperature difference than 40 deg. fahr. less than that of water at a depth of 10 ft., the

[1] Professor of Mechanical Engineering, University of Toronto, Toronto, Canada. Mem. A.S.M.E.

average difference being 3 deg. At the proposed site, water at a depth of 45 ft. was usually only 2 deg. fahr. warmer but rose at times to 5 deg. warmer than at the present site where the depth is 69 ft. There was very definite evidence of sub-surface currents which made an exact prediction about the temperature quite impossible.

To obtain the deeper water involves much difficulty and expense in the majority of cases, because in the larger lakes the bed generally has a gentle slope near the shore. Thus at Cleveland the waterworks crib situated over 10,000 ft. from shore is only in 48½ ft. of water. At Chicago, a crib about 16,000 ft. out is in 34 ft. of water; at Milwaukee, 43 ft. depth is reached at a distance of 4000 ft., while at Toronto, the depth is 69 ft. 2700 ft. from shore at the Island, although elsewhere it is necessary to go out 6000 ft. before reaching a depth of 49 ft. It might not be unreasonable, therefore, to say that an intake to be effective from the author's standpoint would probably extend at least 3000 ft. from shore. The intake line would be exposed to severe weather, and therefore a crib and tunnel of permanent form would need to be constructed.

A station of 100,000 kw. capacity using 2 gal. of cooling water per kilowatt per minute would require 200,000 gal. per min. or about 450 cu. ft. per sec., and this would mean an intake tunnel 12 ft. in diameter if circular. Such a tunnel with proper protecting crib could not be built for less than half a million dollars, and if an increase in the size of the station was contemplated for the future, the intake would have to be made to suit the ultimate capacity, thus making the cost still greater.

It is extremely difficult to design an intake that will not take a large proportion of its supply from above it, only a part being taken from the level of the intake ports, since the intake has a marked tendency to draw from the cone of water reaching from the ports toward the surface. Even in cribs with ports at a depth of 30 ft., the effect on the surface is often quite noticeable, indicating that some surface water is being drawn in. The writer does not agree with the last sentence in Par. 26, as experience shows that water-works intakes have not fulfilled the conditions desired by the author.

Further, the ports or openings in the crib would have to be at least five feet above the lake bed to avoid interference from débris, and it would therefore be necessary to extend the intake into much deeper water than the effective depth at which draft of water was desired. In the case shown by the author in Fig. 3, the intake would have to be placed where the depth of the lake exceeded 80 to 85 ft. to secure the results he desires. It would therefore appear that practical application of the author's scheme could be made only in special cases.

The problem of getting water from the lower surfaces is one that occurs in water-power plants as well as in steam plants. At

the Annual Meeting of the Society in 1924 the writer presented a paper on intakes for power plants,[1] which dealt with the important problem of trying to secure water from the bottom of a river with a view to avoiding surface ice. If one is to avoid taking water from the surface, the maximum velocity at the intake must be extremely low, and even with a designed average velocity of two to three feet per second at the ports, a maximum velocity of six feet per second might be reached at some places, which would cause some surface water to enter an intake 30 ft. below the surface.

If surface water is to be avoided, it is necessary to have the maximum intake velocity extremely low. In some steam power plants this has been neglected, the design giving a low average velocity, without regard to the fact that the velocity is not uniform over the entire surface of the intake, so that the maximum velocity may be much greater than the mean.

M. K. Drewry.[2] The application of the thought developed in Professor Christie's paper to conditions on Lake Michigan may be of interest in connection with his paper. The advisability of drawing condensing water from a considerable depth in Lake Michigan at Milwaukee has been investigated in a summary manner, and depth-temperature readings from other places on the Great Lakes have been obtained.

As a basis of what might be expected at Milwaukee where condensing water is obtained at a considerable distance out in the lake, data obtained from construction and operation of the city-water-supply intake tunnel are submitted. It is thought that the same permanence and reliability of that tunnel would be required in a power-plant intake, and therefore the construction costs and water temperatures obtained should apply. They have been compared with Lakeside Station data and the results tabulated as follows:

	Lakeside Station	Milwaukee Water Works
Type of intake	Shore	Submerged 60 ft.
Capacity, gal. per min.	250,000	150,000
Cost, dollars	$750,000	$1,300,000
Water temperatures, average 1923-1924		
January	40	35.5
February	39	34.5
March	39.5	34.5
April	44	38
May	50	43
June	57	46
July	63.5	50.5
August	65	55.5
September	61.5	52.5
October	55	53
November	48	47
December	41	39.5
Average	50.3	44

[1] Intakes for Power Plants, Trans. A.S.M.E., vol. 46 (1924), p. 1131.

[2] Assistant Chief Engineer of Power Plants, The Milwaukee Electric Railway and Light Company, Milwaukee, Wis. Jun. A.S.M.E.

Average exhaust pressure, in. Hg abs	0.85	0.70
Percentage difference in water rate	...	0.62
Annual saving in coal bill (200,000-kw. station, fuel at 0.3 cent per kw-hr., 55 per cent load factor)	...	$18,000
Justified additional expenditure, 10-yr. repayment	...	$180,000

The cost of the Lakeside intake includes that of all marine works, and tunnels and shafts in the plant. The cost of the water-works intake, embracing shore shaft, lake tunnel, lake shaft, crib, and steel crib used in construction, and including all engineering and inspection, was slightly over 1¼ millions, or half a million dollars in excess of the Lakeside costs. The water-works tunnel extends 5565 ft. out into the lake into 67 ft. depth of water, and is 12 ft. in internal diameter. The intake crib is octagonal, 80 ft. in diameter and 12 ft. in height. The capacity recorded is computed on a velocity basis of 3 ft. per sec. Five years was required for completion of the tunnel and crib.

The "average exhaust pressure" item was derived from condenser-performance standards established at the Lakeside station, and the exhaust pressure corrections applied are those for the 30,000-kw. units installed at Lakeside, assuming 25,000-kw. load.

It appears that a deep-water intake at Milwaukee is not feasible because of the excessive construction costs. In cases where deep water occurs nearer the shore, conclusions may obviously be different.

Data obtained from a Government report [1] and showing temperature-depth relations in ports of the Great Lakes other than at Milwaukee are herewith recorded.

Lake	Michigan	Michigan-Huron	Huron
Place of measurement	Frankfort	Mackinac Straits	Harbor Beach
Date	Aug. 11, 1911		Aug. 5, 1911
Depth, ft.	Temperature, deg. fahr.		
0	64.5	64	68
20	..	64	64.5
40	..	62	60
50	58	..	..
60	..	57	55
80	..	56	54
100	47	55	46.5
120	..	55	45.5
140	..	54.5	44.5
150	46	..	44
200	46	53	..
250	46	..	..
300	45	..	..
400	48	..	..
500	47	..	..
600	48	..	..

THE AUTHOR. Since the idea presented in the paper seems rather new to engineers, they naturally are not yet in a position to discuss it very fully. In regard to the discussions of Mr. Milne and Professor Angus, the author has some additional data on the Toronto

[1] House Document No. 762, 63d Congress, Second Session.

situation. There are a few facts that were not altogether brought out by these two gentlemen. For instance, Professor Angus gave some September readings, showing only a few degrees difference in temperature between the 10-ft. and 45-ft. depths. There happen to be available a number of curves covering a long period of years at Toronto, which indicate for some peculiar reason the lake temperature at the intake to be highest in September. September 6, which is practically the end of the summer in Toronto, had the hottest intake water. Apparently there was at that time some storm or turnover, or something else that mixed the water and caused the heavy current, to which Professor Angus referred.

In view of these additional curves, the very small differences between the temperatures at those levels at that particular time is not surprising. It is probable that if Professor Angus had taken June readings, when the water temperature at the intake is scarcely more than 45 deg. fahr., and when the surface water is quite warm, he would probably have found a greater difference than in that particular case.

It is gratifying to have Professor Angus bring out the point that there must be considerable attention given to the hydraulic features of the intake design in power houses. As engineers we have neglected some of the hydraulic features of the design for intakes and discharges, and it is quite possible that we might reduce materially the curve of pumping head on the circulating pump by considering the proper hydraulic design of the intake and discharges. We shall have gained considerably in reducing the water temperatures, as has been pointed out, by a proper design of intake, and probably will not have so much air in the intakes.

No. 1960

THE RATIONAL DESIGN OF COVERING FOR PIPES CARRYING STEAM UP TO 800 DEG. FAHR.

By W. A. Carter,[1] Detroit, Mich.
and
E. T. Cope,[2] Detroit, Mich.
Associate-Members of the Society

The authors present an analysis of the problem of determining the most economical thickness of pipe covering for steam pipes carrying steam of a total temperature of 500 deg. fahr. and at higher temperatures up to 800 deg. fahr. based on the work of Heilman, Eberle, and others. Two cases are considered, first, in which the total steam temperature is 500 deg. fahr. and a single covering material is used; second, in which the total steam temperature is 700 deg. fahr. and a compound covering, made up of a layer of higher temperature-resisting material next to the pipe and a layer of lower heat-resisting material over it. Curves showing the method of determining the most economical thickness of covering for a 6-in. pipe carrying steam at 500 deg. fahr. and 700 deg. fahr. are included. Equations for the calculations involved are included and the procedure of making the calculations is explained. Three appendices deal with some of the assumptions made in the analysis.

Introduction

ENGINEERING publications known to the authors contain no rational method of determining the correct thickness of covering to be placed on pipes carrying steam and hot water. The recent increase of the total temperature of steam to 700 deg. fahr., and over, makes the problem of accurately estimating and controlling the heat loss from pipes carrying superheated steam more important than ever before.

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[2] Research Dept., The Detroit Edison Co.

Contributed by the Power Division and presented at the Spring Meeting, Milwaukee, Wis., May 18 to 21, 1925, of The American Society of Mechanical Engineers.

2 Many parts of the pipe-covering problem have been reduced to accurate analysis. Heilman[1] produced accurate figures on the loss of heat from bare wrought-iron pipe and the loss of heat from the canvas surface of covered steam pipe. The law governing the flow of heat through thick cylindrical walls, such as the covering on pipes, has been accurately determined.[2] But as far as known to the authors these findings have not previously been incorporated into a comprehensive method for determining the proper thickness in any given case.

Procedure

3 The design of pipe covering for each size of pipe in a large central station which was recently built required the determination of the following items, estimated on the basis of 1 linear foot per year:

a Heat loss from bare pipe
b Heat loss through various thicknesses of covering
c Heat saving due to various thicknesses of covering, (equals $a-b$)
d Gross monetary saving due to various thicknesses of covering, (equals monetary value of heat saving in *c*)
e Fixed charges for various thicknesses of covering
f Net monetary saving due to various thicknesses of covering, (equals $d-e$).

The commercial thickness of covering nearest to that showing the greatest net monetary saving was the one selected as being the most economical.

4 No credit was given to the pipe covering for the improvement in steam consumption of the apparatus using the steam resulting from the increase of steam temperature or steam quality, as the case might be, due to the reduction of heat loss from the covered pipe as compared with that from bare pipe.

Assumptions

5 *Cases Considered.* There were two cases considered. In the first case a single insulating material was used, while in the second case it was considered necessary to use a material of higher heat-resisting properties between the pipe and the same kind of material used in the first case, on account of the excessive steam temperature.

[1] Heat Loss from Bare and Covered Wrought-Iron Pipe at Temperatures up to 800 Deg. Fahr., R. H. Heilman, Trans. A.S.M.E., vol. 44, 1922, p. 299.

[2] Mathematical Theory of Heat Conduction (Par. 31), Ingersoll & Zobel, 1913, Ginn & Company. Introduction to the Mathematical Theory of the Conduction of Heat in Solids, H. S. Carslaw (p. 114, par. 54), 1921, MacMillan & Company.

6 This analysis considers only a 6-in. pipe surrounded by air at 80 deg. fahr. The cost of the heat in the steam was taken as 35 cents per million B.t.u. In Case 1, the total temperature of the superheated steam was 500 deg. fahr., while in Case 2 it was 700 deg. fahr.

7 *Temperature of Outside Surface of Pipe.* It is generally understood that the temperature on the outside surface of a bare pipe carrying superheated steam is lower than that of the steam, and that the greater the amount of thermal insulation that is applied to the pipe the more closely will the temperature of the outside surface of the pipe approach that of the steam.[1] This is due to the decrease in the amount of heat flowing from the steam to the outside surface of the pipe, after the application of the covering.

8 For lack of sufficient data to fix the temperature of the outside surface of the bare pipe, and inasmuch as the ultimate solution of the problem would not be influenced by the amount of temperature drop between the steam in the pipe and the outside surface of the pipe, the logic of which is set forth in Appendix No. 1, it was assumed that the outside surface of the bare pipe was at the same temperature as the superheated steam in the pipe.

9 When calculating the heat loss from covered pipe it was assumed that the temperature of the outside surface of the pipe was the same as that of the superheated steam. The insignificance of the temperature difference between these two points in a well-covered pipe was borne out by tests made at a large power station to determine the accuracy of the industrial-type thermometer. Comparison was made with a calibrated thermocouple welded into the bottom of a standard thermometer well of the finned type designed for superheated steam. A second calibrated thermocouple was peened into a small drill hole in the *outside* of the pipe. These two thermocouples were in the same foot of length of the pipe which was amply insulated. Every precaution was taken to prevent conduction of heat away from the hot junction of the latter thermocouple. Its lead wires were insulated from the pipe and were wound around the pipe one complete turn before being led out through the pipe covering. The cold junctions were immersed in an ice bath. In short, every known precaution was taken to insure the highest degree of accuracy. The log of a representative

[1] In the case of a pipe passing wet or dry saturated steam the temperature of the outside surface of the pipe is practically the same as that of the steam, regardless of whether it is bare or covered. This is due to the presence of a film of condensate at the same temperature as the steam on the inner surface of the pipe, and of the low resistance to heat flow from the steam through this water and the pipe wall, which results in a small temperature differential between the steam and the outside surface of the pipe for the amount of heat flowing between those two points.

half-hour of readings is shown in Fig. 1. At no time was there a difference of more than three or four degrees between the two thermocouples.

10 Further proof that the temperature difference between the superheated steam in an insulated pipe and the outside surface of the pipe is small, is to be found in the graphical solution of the example given in Appendix No. 2, wherein the temperature drop from superheated steam to the outside of the pipe surface necessary to produce the heat flow through the covering and into the air was calculated by Eberle's formula.[1]

11 *Temperature of Inside Surface of Covering.* It was also assumed that the temperature of the inside surface of the pipe covering was equal to the temperature of the pipe. There is

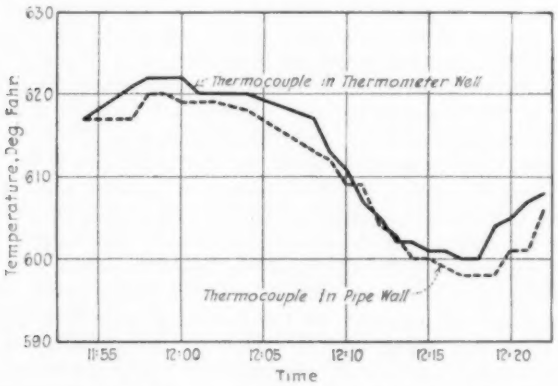

FIG. 1 TEMPERATURE READINGS TAKEN ON A COVERED PIPE CARRYING SUPERHEATED STEAM

doubtlessly a drop in temperature from the outside surface of the pipe to the inside surface of the covering. Assuming the temperature of the inside of the covering equal to that of the pipe would result in a greater temperature difference between the inside and outside surfaces of the covering than would obtain if the temperature drop mentioned above were considered.

12 This consequently gives the greatest possible heat flow through the covering. Therefore the error, if any, is on the safe side.

13 In the following paragraphs are given the formulas used in making the calculations for each step in the procedure which has already been outlined, and a solution of Case 1. Immediately following that the solution for Case 2 is given.

[1] Mitt. über Forschungsarbeiten auf dem Geb. des Ing.. Heft 78. 1909.

Heat Loss from Bare Pipe

14 It was necessary first to determine the loss of heat from the bare pipe. Heilman has reported results of experiments, conducted at the Mellon Institute of Industrial Research, to determine the rate of heat transfer from bare wrought-iron pipe of certain sizes to air at various temperature differences. The heat-transfer rate was expressed in B.t.u. per hour per linear foot of bare iron pipe per deg. fahr. temperature difference between the pipe and the air. Fig. 2 shows the values of the heat loss in B.t.u.

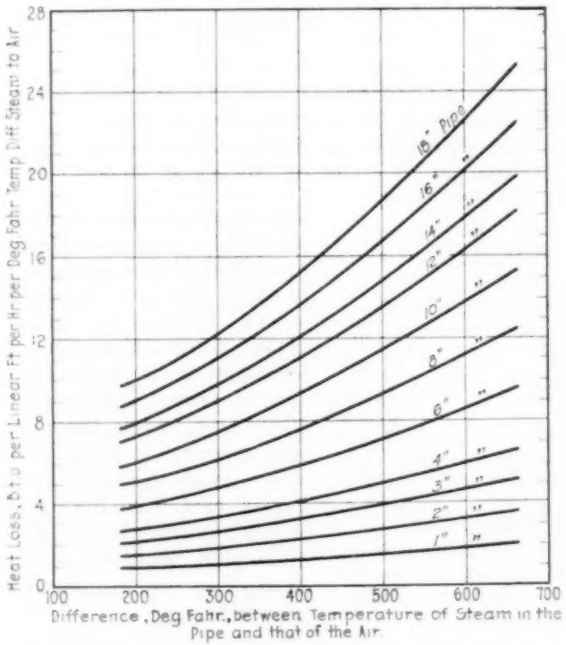

Fig. 2 Rate of Heat Transfer from Wrought-Iron Pipe to Air, at Various Temperature Differences

per hour per linear foot of pipe per deg. fahr. temperature difference between the steam and the air, assuming that the steam and the pipe wall are at the same temperature.

Heat Loss from Covered Pipe: Case 1

15 All of the heat that escapes from the surface of the pipe covering into the surrounding air must have passed through the covering. The magnitude of this heat loss was derived from the calculated heat loss from the outside of the covering and the heat conducted through the covering for various assumed temperatures

of the canvas surface. A graphical method was used to determine the heat loss that would result in a common temperature for the canvas surface.

16 Heilman derived an empirical expression, which was based on experimental data, for the loss of heat from the canvas surface of insulated pipes. This expression was applicable to cylindrical surfaces up to 24 in. in diameter. His formula was

$$h_1 = \frac{\frac{564}{D^{0.19}}(t_c - t_a)}{272.5 - (t_c - t_a)} \quad \ldots\ldots\ldots [1]$$

in which

h_1 = B.t.u. loss per hr. per sq. ft. of canvas surface exposed
D = diameter of the canvas surface in inches
t_c = temperature of the canvas surface in deg. fahr.
t_a = temperature of the surrounding air in deg. fahr.

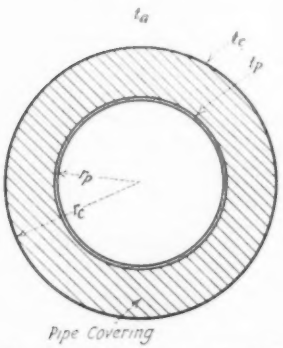

FIG. 3 SECTION THROUGH PIPE SHOWING SINGLE LAYER OF COVERING

For convenience this expression was changed to read in B.t.u. loss per linear foot of pipe per hour instead of B.t.u. loss per square foot of canvas surface per hour. Few persons have a clear conception of the extent or dimensions of a square foot of canvas surface on, say, a 10-in. pipe, having three inches of pipe covering. It is not at all difficult to grasp the significance of the loss of heat from the canvas surface of a one-foot length of the same 10-in. pipe. When the equation was changed to read per linear foot of pipe it became

$$h_2 = \frac{\frac{564}{D^{0.19}}(t_c - t_a)}{272.5 - (t_c - t_a)} \times \frac{\pi D}{12} \quad \ldots\ldots [2]$$

in which h_2 = the B.t.u. loss per linear foot of pipe. This equation was simplified to read

$$h_2 = \frac{147.6 D^{0.81}(t_c - t_a)}{272.5 - (t_c - t_a)} \quad \ldots\ldots\ldots [3]$$

17 The rate at which the heat passes through the layer of pipe covering conforms to the law for the passage of heat through a thick cylindrical wall. It was assumed that the pipe covering was uniform in thickness and in composition. The rate of conduc-

tion of heat through the layer of pipe covering, shown in section in Fig. 3, was expressed by the equation [1]

$$h_3 = \frac{K(t_p - t_c)}{r \log_e \frac{r_c}{r_p}} \quad \ldots\ldots\ldots [4]$$

in which

h_3 = B.t.u. per hr. per sq. ft. (of cylindrical surface with radius r) flowing from inner surface to outer surface of pipe covering

K = mean absolute thermal conductivity of the insulating material expressed in B.t.u. per sq. ft. per hr. per in. of thickness per deg. fahr. temperature difference between the hot and cold surfaces. K is not a constant but changes with the mean temperature of the material

t_p = temperature of the inside surface of the layer of pipe covering in deg. fahr. This is taken the same as the temperature of the steam

t_c = temperature of the outside surface of the layer of pipe covering in deg. fahr.

r = the radius, in inches, of the surface used as a basis for h_3. In this analysis, r was the radius of the outside surface of the covering = r_c

r_c = the radius, in inches, of the outside surface of the pipe covering

r_p = the radius, in inches, of the inside surface of the pipe covering. This was taken equal to the outside radius of the pipe.

Here also it was desirable to change this equation to read in terms of one linear foot of the pipe. When this was done, the equation became

$$h_4 = \frac{K(t_p - t_c)}{r_c \log_e \frac{r_c}{r_p}} \times \frac{2\pi r_c}{12} \quad \ldots\ldots\ldots [5]$$

where h_4 = the B.t.u. passing through the layer of pipe covering per hour per linear foot of pipe. Reduced to its simplest form this became

$$h_4 = \frac{0.524K(t_p - t_c)}{\log_e \frac{r_c}{r_p}} \quad \ldots\ldots\ldots [6]$$

[1] Introduction to the Mathematical Theory of the Conduction of Heat in Solids, H. S. Carslaw (p. 114, par. 54), 1921, MacMillan & Co. Mathematical Theory of Heat Conduction, Ingersoll & Zobel (par. 31), 1913, Ginn & Co.

18 Assuming that thermal equilibrium has been established, the heat loss from the canvas covering per linear foot of pipe must equal the heat conducted through the covering per linear foot of pipe. Consequently, Equations [3] and [6] become equal to each other; that is,

$$\text{Heat loss} = \begin{bmatrix}\text{Heat loss from}\\ \text{canvas surface}\end{bmatrix} = \begin{bmatrix}\text{Heat conducted through the}\\ \text{covering}\end{bmatrix}$$
$$= h_2 = h_4$$

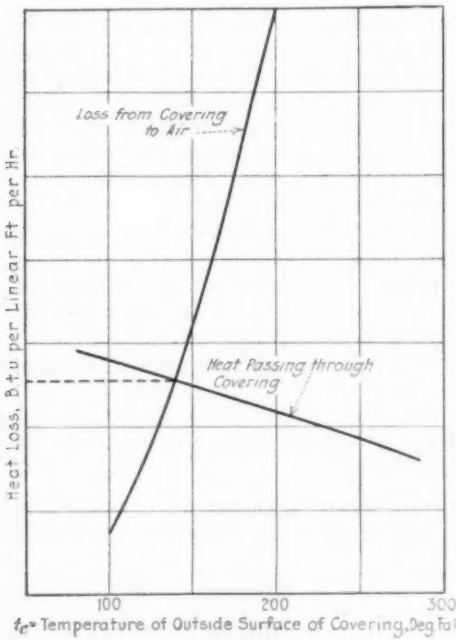
Loss from Covering to Air
Heat Passing through Covering
Heat Loss, B.t.u. per Linear Ft per Hr
100
200
300
t_c= Temperature of Outside Surface of Covering, Deg Fahr

FIG. 4 HEAT LOSS THROUGH 1½-IN. COVERING ON A 6-IN. PIPE

(See footnote for Par. 18)

A graphical method was used to solve Equations [3] and [6] simultaneously. In Equation [3] the known quantities were t_a and D which equals $2\times$ (r_p plus assumed thickness of covering). Values of h_2 were calculated for assumed values of t_c which were 100, 150, and 200 deg. fahr. The curve marked "Loss from Covering to Air," Fig. 4, is a plot of the results.[1]

[1] The actual values of heat loss in this and subsequent figures are omitted, as they are not essential to demonstrate this method of solution.

19 In Equation [6] the known quantities were K taken from a curve for the average temperatures of the covering as fixed by t_p and the three assumed values of t_c, t_p, r_p, and r_c which equals (r_p plus assumed thickness of covering). Again various values for t_c were assumed and this equation was solved for h_4. These values were plotted on the same sheet as the loss-from-covering-to-air

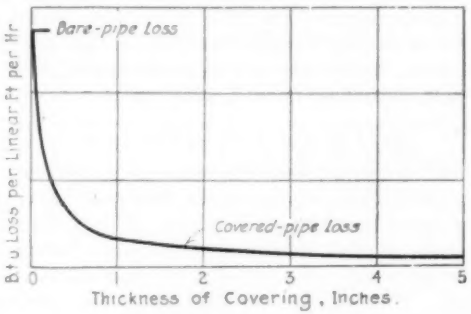

FIG. 5 B.T.U. LOSS FROM A 6-IN. PIPE CARRYING STEAM AT 500 DEG. FAHR.

(See footnote for Par. 18)

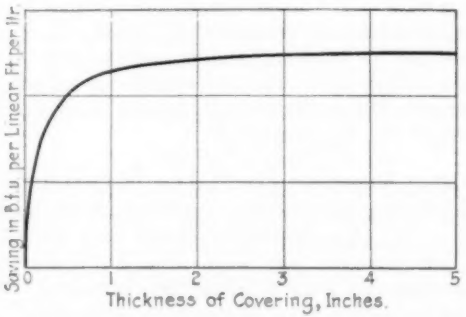

FIG. 6 B.T.U. SAVING RESULTING FROM VARIOUS THICKNESSES OF PIPE COVERING ON A 6-IN. PIPE CARRYING STEAM AT 500 DEG. FAHR.

(See footnote for Par. 18)

curve, Fig. 4. The values for t_c were taken as 70, 200, and 300 deg. fahr. This curve is marked " Heat Passing through Covering." The point of intersection of these two curves indicates a condition of thermal equilibrium for the assumed thickness of covering. It is then possible to pick off the loss in B.t.u. per linear foot of pipe by taking the value of the ordinate of this intersection. The abscissa of this point of intersection indicates the actual temperature of the outside of the covering.

20 By selecting several other thicknesses of pipe covering for the chosen pipe and the chosen temperature of steam, and carrying out the above analysis, the B.t.u. loss per linear foot of pipe was obtained for several thicknesses of covering.

21 By plotting these values of B.t.u. loss per linear foot of pipe having different thicknesses of covering against the thickness of the covering, Fig. 5 was obtained. The heat loss per linear foot of the bare pipe as obtained from Fig. 2 also was plotted in Fig. 5. This curve shows the loss of heat from the 6-in. pipe carrying steam at 500 deg. fahr., both bare and when protected with the particular covering considered in thicknesses up to 5 in.

Heat Saving Due to Covering

22 The difference between the bare-pipe loss and the loss through any particular thickness of the pipe covering is the saving in B.t.u. per linear foot of pipe per hour. This saving is effected by the presence of that thickness of the pipe covering. Fig. 6 shows the heat-saving curve for a 6-in. pipe carrying steam at 500 deg. fahr. The ordinates of this curve were obtained from Fig. 5.

23 There are several striking characteristics of the curve in Fig. 6. It approaches tangency to a horizontal line at its right-hand end. This means that beyond a fairly definite thickness of covering on a pipe of any given size carrying steam at any given temperature, an additional thickness of covering is of very little, if any, value. At its left-hand end the curve rises very rapidly and has a very sharp knee. The first quarter of an inch thickness of pipe covering is of more value in saving heat than any additional inch thickness of the same material.

Gross Monetary Saving Due to Covering

24 The gross monetary saving due to the various thicknesses of covering is the monetary value of the heat saving. In this study, the cost of steam was taken from the operating records of a certain large plant for a recent annual period. This plant showed a cost per million B.t.u. in the steam of 35 cents. Any other reasonable cost would have been equally satisfactory. The gross-monetary-saving curve for the case under consideration is shown in Fig. 7. The effect of a different cost per million B.t.u. upon the most economical thickness is discussed in Appendix No. 3.

Fixed Charges

25 The fixed charges on the covering, including maintenance, depreciation, interest, and taxes, were taken as 20 per cent of the cost of the covering in place. The cost per linear foot of covering of any particular thickness includes the cost of the materials, freight charges, and the cost of labor to place the covering on the pipe. The cost of labor for installing the covering was an average

estimated cost obtained from two nationally known companies that manufacture pipe covering and apply it under contract.

26 The fixed-charges curve was plotted as shown in Fig. 7. As the thickness of covering increased the slope of this curve increased. It was really not quite as smooth a curve as shown, because commercial considerations required that definite steps of increase of covering thickness be made. The error introduced by considering this smooth curve was small.

Net Monetary Saving Due to Covering

27 The net-monetary-saving curve shown in Fig. 7 is the difference between the gross-monetary-saving curve and the fixed-

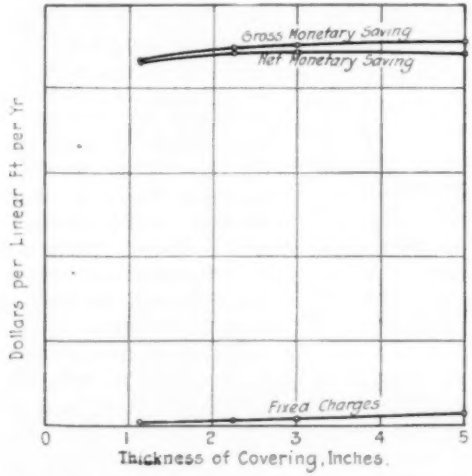

FIG. 7 CURVES USED IN SELECTING THE MOST ECONOMICAL THICKNESS OF COVERING FOR A 6-IN. PIPE CARRYING STEAM AT 500 DEG. FAHR.
(See footnote for Par. 18)

charges curve. The net-monetary-saving curve contains the answer of this problem.

28 Certain characteristics of this curve are rather striking. The curve is very flat. This indicates that there is a rather wide latitude in the choice of the proper thickness of covering. Fairly wide variation in thickness causes little difference in net monetary saving.

29 Steam-pipe covering is supplied in definite commercial thicknesses. Some are made up of one layer and others of two layers. It is therefore possible to select the commercial thickness which most nearly fits the conclusions shown in Fig. 7. A thickness of 3 in. seemed to be about the best for the case considered. A covering made up of two layers of 1½ in. thickness each was the one chosen for the particular material considered.

HEAT LOSS FROM PIPES HAVING COMPOUND COVERING: CASE 2

30 The analysis just concluded has assumed that a single material is used as pipe covering. In the case just discussed 85 per cent magnesia was considered on pipes carrying steam up to 500 deg. fahr. For pipes carrying steam at temperatures in excess of 500 deg. fahr., the covering was made up of a protective layer next to the pipe and 85 per cent magnesia over this protective layer.

31 Some investigations have disclosed that 85 per cent magnesia insulation undergoes a change when heated continuously at a temperature ranging from 500 to 600 deg. fahr., which change weakens the material mechanically.

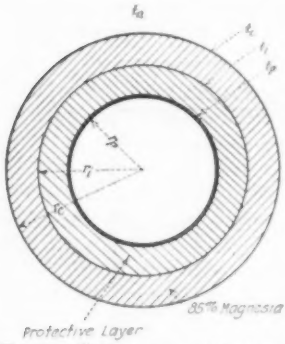

FIG. 8 SECTION THROUGH PIPE SHOWING COMPOUND COVERING

32 In consequence of these observations, it was decided arbitrarily to set 500 deg. fahr. as a maximum temperature to which the inside of the layer of 85 per cent magnesia covering would be subjected. A material which would not show serious loss of mechanical strength at a temperature higher than 500 deg. fahr. was selected to be used next to the pipe.

33 When such a combination of pipe covering, made of layers of two different materials as shown in Fig. 8, was used, the design method was changed only in that the two materials would not ordinarily have the same thermal-conductivity values. The equation for the thermal equilibrium was expanded to include the passage of heat through a second layer of insulating material. This equation for a compound covering became

$$\text{Heat loss} = \begin{bmatrix}\text{Heat loss}\\ \text{from can-}\\ \text{vas cover}\end{bmatrix} = \begin{bmatrix}\text{Heat conducted}\\ \text{through outer}\\ \text{layer}\end{bmatrix} = \begin{bmatrix}\text{Heat conducted}\\ \text{through pro-}\\ \text{tective layer}\end{bmatrix}$$

$$= h_2 = h_5 = h_6$$

$$h_2 = \frac{147.6D^{0.81}(t_c - t_a)}{272.5 - (t_c - t_a)} \quad \ldots\ldots\ldots\ldots [3]$$

$$h_5 = \frac{0.524K_a(t_i - t_c)}{\log_e \frac{r_c}{r_i}} \quad \ldots\ldots\ldots\ldots [7]$$

$$h_6 = \frac{0.524K_b(t_p - t_i)}{\log_e \frac{r_i}{r_p}} \quad \ldots\ldots\ldots\ldots [8]$$

The symbols are identical with those used in Case 1, with the addition of K_a, K_b, r_i, and t_i.

h_5 = B.t.u. conducted through the outer layer per hour per linear foot of pipe

h_6 = B.t.u. conducted through the protective layer per hour per linear foot of pipe

K_a = mean absolute thermal conductivity of the outside layer. K_a is expressed in B.t.u. per sq. ft. per hr. per in. thickness per deg. fahr. temperature difference between the inside and the outside of this layer of material

K_b = a similar value to K_a for the protective layer of the compound covering

r_i = the radius in inches of the outside surface of the layer of protective material

t_i = the temperature of the outside of the layer of protective material. It was assumed that there was no drop in temperature between the outside surface of the protective layer and the inside surface of the outside layer.

34 The solution of this set of equations was carried out as follows: The loss of heat from the canvas surface was calculated from Equation [3]. Values of t_c were assumed after D had been taken at some convenient value. The total covering thickness on a 6-in. pipe was taken equal to 4 in. Therefore D equaled the outside diameter of the pipe plus 8 in.; t_c was assumed as 80, 100, and 150 deg. fahr., and the equation was solved for each value of t_c. Values of h_2 were thus obtained for these values of t_c. The loss-from-canvas-surface curve was plotted between h_2 and t_c. This is shown on the right-hand side of Fig. 9 and is marked "Loss from Canvas Surface."

35 Next, the heat passing through the outside layer of the covering was computed by the use of Equation [7]. In this equation K_a was obtained from the thermal-conductivity curve of the material for the temperature $\frac{(t_i+t_c)}{2}$. The temperature of the inner surface of the outer layer of covering, t_i, was taken as 500 deg. fahr., the assumed permissible maximum temperature. The quantity r_i was taken at several convenient thicknesses of the inside layer of the covering. Values of r_i equal to 3.3125, 4.3125, and 5.3125 in. were obtained by assuming thicknesses of protective covering equal to 0, 1, and 2 in., respectively. Values of t_c of 100, 200, and 300 deg. fahr., were assumed for each value of r_i. The various values of h_5 were calculated and a curve plotted between h_5 and t_c for each value of r_i. This gave the family of curves marked "Loss through Outer Covering," Fig. 9. These curves intersected the loss-from-canvas-surface curves at

values of heat loss and t_c representing thermal-equilibrium conditions.

36 Next, a curve plotting the ordinates of these points of intersection as ordinates and the corresponding radii (r_i) as

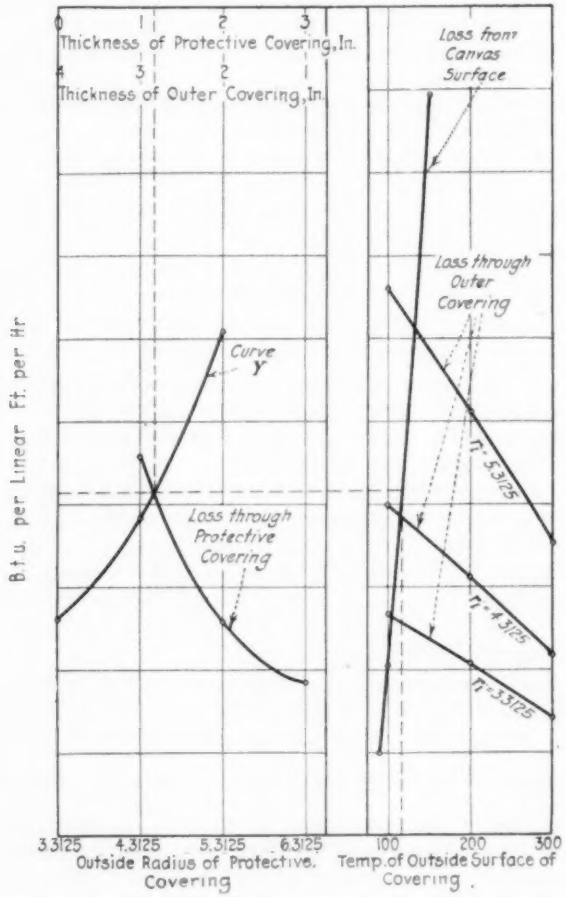

FIG. 9 DETERMINATION OF HEAT LOSS PER LINEAR FOOT FROM A 6-IN. PIPE HAVING 4 IN. OF COMPOUND COVERING AND CARRYING STEAM AT 700 DEG. FAHR.

(See footnote for Par. 18)

abscissas was laid out on the left-hand side of the sheet. This is curve Y, Fig. 9.

37 Then, values of heat passing through the layer of protective material were calculated by means of Equation [8]. In this equa-

tion, r_i was the only unknown since t_p was assumed equal to the temperature of the steam and t_i equaled 500 deg. fahr., by assumption. Substituting in Equation [8] the same values of r_i used in Equation [7], the various values of h_6 were calculated. Then the curve which is marked "Loss through Protective Covering" was plotted on the left-hand side of Fig. 9 between the values of h_6 just calculated as ordinates and the corresponding radii (r_i) as abscissas.

38 Thermal equilibrium of the system between the steam in the pipe and the surrounding air occurs, for the conditions assumed, when the loss in B.t.u. per linear foot of pipe per hour equals that represented by the ordinate of the intersection of curve Y and the loss-through-protective-covering curve. If a horizontal line be drawn toward the right from this point of intersection until it intersects the curve marked "Loss from Canvas Surface," the temperature of the outside surface of covering corresponding to this point is the temperature of the outside surface of the covering for thermal-equilibrium conditions. The ordinate of this point of intersection is the heat loss per linear foot of pipe under the same conditions.

39 At the top of Fig. 9, toward the left, are two scales, one marked "Thickness of Protective Covering," the other marked "Thickness of Outer Covering." The total covering thickness used in this solution was 4 in. This may be made up of 1 in. of protective layer and 3 in. of outer layer, or 2 in. of each, or 3 in. of protective layer and 1 in. of outer layer. A 1-in. protective layer makes an outside radius of protective covering on a 6-in. pipe equal to 4.3125 in. On the thickness-of-protective-layer scale at the top of Fig. 9, 1, 2, and 3 correspond with 4.3125, 5.3125, and 6.3125, respectively, on the outside-radius-of-protective-covering scale at the bottom of Fig. 9. A vertical line drawn upward from the intersection of curve Y and the loss-through-protective-covering curve intersects the thickness-of-protective-covering scale at $1\frac{3}{16}$ in. The thickness of the outer layer is $2\frac{13}{16}$ in. (equals $4-1\frac{3}{16}$). These dimensions satisfy the condition imposed, namely, that the temperature of the inside surface of the outer layer shall be 500 deg. fahr.

40 All that it was necessary to do then was to plot three or four of these curves for different total thicknesses of covering, take off the loss of heat in B.t.u. per linear foot of pipe and plot the heat-saving curves for various total covering thicknesses as shown in Fig. 6. From this point the procedure was the same as is described in the early part of this paper. When the analysis was completed it was necessary to select the thicknesses for both the protective layer and the outside layer that were obtainable in commercial sizes.

41 If the thickness of the protective layer obtained was not a commercial standard, a greater rather than a lesser thickness than

that dictated by the analysis was chosen, thereby subjecting the 85 per cent magnesia to a temperature lower than the maximum arbitrarily set.

42 It is obviously out of place in such a paper as this to go into a discussion of actual materials that lend themselves readily for use in the protective layer. It is equally obvious that exact values of thicknesses of pipe covering cannot be stated because different brands of these materials have different thermal conductivities and are sold at different prices.

APPENDIX NO. 1

DEMONSTRATION THAT DROP IN TEMPERATURE FROM STEAM TO OUTSIDE OF BARE PIPE DOES NOT AFFECT SELECTION OF MOST ECONOMICAL THICKNESS OF PIPE COVERING

43 It has been stated in Par. 22 that the heat saving produced by the covering on a steam pipe is the difference between the loss of heat from the bare pipe and the loss of heat from the same pipe when insulated. The magnitude of the heat loss from bare pipe depends on the temperature difference between the pipe and the surrounding air. This is axiomatic.

44 There has been much discussion among engineers and physicists in regard to the difference in temperature between the superheated steam flowing in a pipe and that of the outside surface of the pipe.

45 B. N. Broido[1] shows differences in temperature as high as 100 deg. fahr. between the superheated steam and the outside of the pipe when the pipe is bare.

46 It is only necessary to consider two cases to see that the difference in bare-pipe loss with different assumptions of pipe temperature does not change the most economical covering thickness on a net-monetary-saving basis. In Fig. 10, curve 1 is the annual gross monetary saving resulting from different thicknesses of covering when the bare pipe wall temperature is the same as the steam, namely, 700 deg. fahr. Curve 2 is the annual net monetary saving produced by different covering thicknesses for the same conditions. This curve is derived from curve 1 by deducting the fixed charges plotted at bottom of Fig. 10. Curve 3 is the annual gross monetary saving for different thickness of covering when the temperature of the pipe used in calculating the bare-pipe loss is 100 deg. fahr. lower than the steam temperature. Curve 4 is the annual-net-monetary-saving curve for the latter conditions.

47 The portions of curves 2 and 4, which are shown, are identical except that the ordinates of 2 are a fixed amount greater than those of curve 4. The high point on curve 2 occurs at the same covering thickness as that of curve 4 which proves that the most economical thickness on a net monetary saving basis is the same in the two cases. This in turn shows that in calculating the loss of heat from a bare pipe

[1] High-Temperature and High-Pressure Steam Lines, B. N. Broido, Trans. A.S.M.E., vol. 44, 1922, p. 1199.

carrying superheated steam, the temperature of the pipe may be taken the same as the temperature of the steam.

48 It is true that the actual value of net monetary saving is affected, but the most economical thickness of covering is not changed.

APPENDIX NO. 2

TEMPERATURE DIFFERENCE BETWEEN SUPERHEATED STEAM AND THE OUTSIDE SURFACE OF THE PIPE IN AN INSULATED SUPERHEATED-STEAM PIPE

49 In order to show the magnitude of the drop in temperature from the superheated steam to the outside surface of the pipe necessary to cause the escaping heat to flow, a complete analysis was made of the

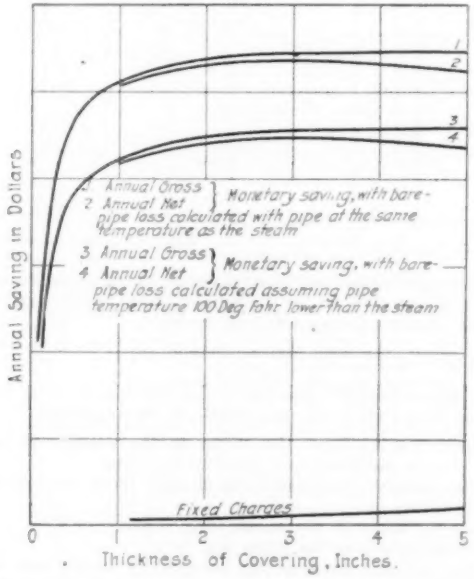

FIG. 10 EFFECT OF DIFFERENT BARE-PIPE TEMPERATURES ON MOST ECONOMICAL THICKNESS OF PIPE COVERING

(See footnote for Par. 18)

heat flow from the superheated steam in the pipe, through the pipe wall, through the covering, into the air.

50 The elements of heat flow in this analysis are as follows:

$$\text{Heat loss} = \begin{bmatrix}\text{Heat escaping}\\ \text{from canvas}\\ \text{covering to}\\ \text{the air}\end{bmatrix} = \begin{bmatrix}\text{Heat passing}\\ \text{through the}\\ \text{covering}\end{bmatrix} = \begin{bmatrix}\text{Heat flowing from}\\ \text{the superheated}\\ \text{steam to the out-}\\ \text{side of the pipe}\end{bmatrix} \quad [9]$$

The first, second, and third members of this expression have been discussed in the earlier part of this paper. The second member is given a value in Equation [3], the third member in Equation [6], and the fourth member is given a value as follows:

$$h_7 = \frac{G\pi(t_s' - t_p')}{\frac{1}{d_1'} \times \frac{1}{a_1} + \frac{1}{2\lambda} \log_e \frac{d_2'}{d_1'}} \quad \ldots\ldots\ldots [10]$$

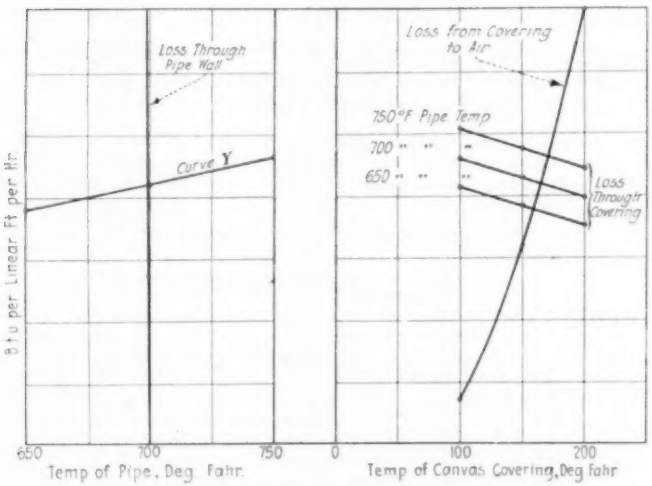

FIG. 11 DETERMINATION OF TEMPERATURE OF OUTSIDE SURFACE OF A 6-IN. PIPE HAVING 1¼ IN. OF COVERING, CARRYING STEAM AT 420 LB. PER SQ. IN. ABSOLUTE AND 700 DEG. FAHR.

(See footnote for Par. 18)

Equation [10] is Eberle's expression for the loss of heat from superheated steam in a pipe to the pipe wall. In this equation

h_7 = heat flowing from the steam to the outside of the pipe in B.t.u. per hour per linear foot of pipe
G = the constant necessary to change from metric to English units
t_s' = temperature of the steam, deg. cent.
t_p' = temperature of the pipe, deg. cent.
λ = the thermal conductivity of steel in kg-cal. per hr., per sq. m., per meter thickness, per deg. cent.
r_1' = inside radius of the pipe in meters
r_p' = outside radius of the pipe in meters.
a_1 = heat-transfer rate from the steam to the inside surface of the pipe in kg-cal. per hr. per sq. m., per deg. cent. temperature difference between the steam and the inside of the pipe.

The value of a_1 is taken from Nusselt's [1] work and equals

$$a_1 = \frac{14.31(wpc_p)^{0.786}\lambda_t^{0.214}}{d_1^{0.186}p^{0.054}} \quad \ldots\ldots\ldots [11]$$

[1] Warmeübergang in Rohrleitungen, Z. V. D. I., Oct. 23, 1909.

in which

w = mean velocity of the steam in meters per second
ρ = density of the steam in kg. per cu. m. at the pressure and temperature given
l = length of pipe in meters
c_p = specific heat of the steam in kg-cal. per kg. at the given pressure and temperature
λ_t = thermal conductivity of the steam at the given temperature and pressure.

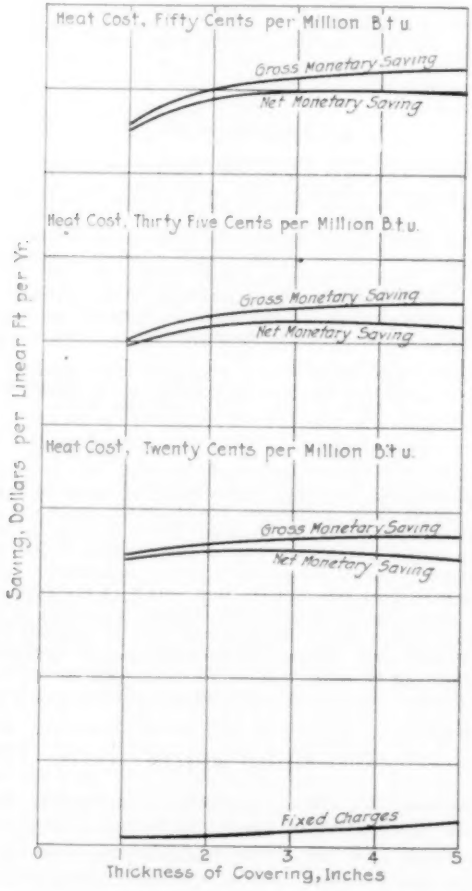

FIG. 12 EFFECT OF DIFFERENT COSTS OF HEAT ON MOST ECONOMICAL THICKNESS OF COVERING

(See footnote for Par. 18)

51 The conditions assumed were:

Steam pressure, 420 lb. per sq. in. abs.
Steam temperature, 700 deg. fahr.
Air temperature, 80 deg. fahr.
Pipe diameter, 6 in., O. D.
Length of pipe, 250 ft.

52 Fig. 11 shows the graphical solution of Equation [9], which is similar to the solution shown in Par. 34. The intersection of curve Y and the curve marked "Loss Through Pipe Wall" shows the actual temperature of the outside surface of the pipe.

53 The temperature difference between the steam and the outside of the pipe for the conditions assumed is about 1 deg. fahr. Even if the temperature difference had been 5, or even 10 deg. fahr., the heat loss per unit time per unit length pipe would not have changed materially.

APPENDIX NO. 3

THE EFFECT OF THE COST OF HEAT IN THE STEAM ON THE MOST ECONOMICAL COVERING THICKNESS

54 Fig. 12 shows the effect of different heat costs on the most economical thickness on the basis of net monetary saving of steam-pipe covering. Heat costs were taken at 20, 35, and 50 cents per million B.t.u. in the steam above feedwater temperature. The same cost of covering was used in each case. The highest points in the net-monetary-saving curves occur at the following covering thicknesses:

Heat cost per million B.t.u., cents	Thickness that produces the greatest net monetary savings, inches
20	2¾
35	3
50	3¼

It is entirely possible that the commercial thicknesses of covering that most nearly meet these conditions would be the same for heat at 20 cents per million B.t.u. as at 50 cents. There is very little difference between the actual net saving over a considerable range of covering thickness in each case.

55 It is true that the cost of fuel which is directly reflected in the cost of steam has some effect on the best thickness of pipe covering, but this effect is small. In the cases cited it is about half an inch difference in thickness. When the final selection of commercial thicknesses of covering is made, the effect of the difference in thickness due to a difference in fuel cost may entirely disappear.

DISCUSSION

R. H. HEILMAN.[1] The importance of accurately determining the economical thickness of insulation to apply to a superheated steam line cannot be too strongly impressed upon the engineer, as there are many steam lines in operation today which are not sufficiently insulated. The fact that a 4-in. thickness of pipe

[1] Senior Industrial Fellow, Mellon Institute of Industrial Research, Pittsburgh, Pa. Assoc-Mem. A.S.M.E.

covering when applied to a 12-in. pipe operating at a temperature of 750 deg. fahr. will pay for itself in a period of 1½ months with steam at only 35 cents per 1,000,000 B.t.u., is conclusive evidence of the importance of insulating all exposed surfaces in accordance with a scientific standard of thickness for pressure.

Several methods can be used in determining the economical thickness of pipe covering. The method used by the Mellon Institute is briefly as follows:

The heat loss through various thicknesses of covering is calculated, and the annual operating charges are determined by multiplying the heat loss so obtained by the cost of the equivalent steam loss per million B.t.u. The annual fixed charges for various thicknesses of covering are then determined, and the annual fixed charges and annual operating charges are plotted on the same sheet as a function of the thickness of the covering.

It is then an easy matter to determine that thickness of covering which gives the minimum annual operating expense either arithmetically or graphically. It is determined arithmetically by measuring the value of heat loss from the annual fixed-charge curve and the annual operating-charge curve at some fixed thickness of covering; adding the two charges together gives the total annual operating charges. The thickness of covering which gives the minimum total annual operating charges is the covering to use. The proper thickness can be determined graphically by locating the thickness at which the tangent to the annual operating-charge curve and the tangent to the annual fixed-charge curve are parallel. At that point, the rate of change of each curve would be the same and the total operating cost would be a minimum. This method is perhaps somewhat easier to employ than the method used by the authors, as it is unnecessary to consider the loss from the bare pipe.

The actual calculation of the heat loss through the pipe coverings, and especially through compound sections, can be somewhat simplified by using the method illustrated by the writer in *Mechanical Engineering*.[1] The procedure used for single-thickness coverings is first to make an assumption of the outer-surface temperature. Then from the conductivity curve the conductivity of the material is determined at the mean of the assumed outer-surface temperature and the inner-surface temperature. The loss through the covering should then be calculated for the assumed outer-surface temperature and this loss should be checked against the loss as obtained from the surface loss equation. If the two losses do not check, the assumed outer-surface temperature must be changed according to the indications of the calculation and the process repeated until the two losses or the outer-surface temperatures check. The method used for compound sections is essen-

[1] *Mechanical Engineering*, vol. 46, no. 10, Oct., 1924, p. 602.

tially the same with the exception that the mean temperature has to be determined for each material, which increases the calculations involved as the number of different layers of material are increased. At first this may seem to be a tedious method, but with the aid of a slide rule one will find that the loss through single and compound sections can be accurately calculated in a very short time. This method eliminates the necessity of drawing any curves.

In regard to the temperature drop from the outside surface of the pipe to the inside surface of the pipe covering, a large number of tests conducted at Mellon Institute indicated that this drop averaged from approximately 5 to 15 deg. fahr., depending upon the pipe size, thickness of covering, temperature, etc. The authors are justified, therefore, in their statement that there is little error in assuming the temperature of the inner surface of the covering to be the same as the temperature of the pipe.

In Par. 23, the authors state that the first quarter of an inch thickness of pipe covering is of more value in saving heat than any additional inch thickness of the same material. While this is absolutely true, the writer would like to caution anyone from assuming that it would not pay to put more than that amount of insulation on a steam line, as it would be very uneconomical to insulate with only $\frac{1}{4}$ in. of material.

G. D. BAGLEY.[1] This paper represents a distinct advance in the economic consideration of the application of pipe coverings for the purpose of reducing heat losses, in that it extends the work already done by others to the higher temperatures which are required by present-day conditions, and also in the consideration of compound coverings which have been made necessary by the higher temperatures used.

A similar method for the determination of the proper thickness of coverings to be used under various conditions of pipe temperature and steam cost was developed in the course of an investigation at the Mellon Institute of Industrial Research in Pittsburgh, in 1918, and published in a paper presented by the writer at the Annual Meeting of the Society in 1918.[2] The method used was basically the same but instead of plotting "net monetary saving" against varying thickness of covering, the operating cost, which consists of the fixed charges plus the value of the heat lost after installation of the covering, was plotted against the thickness. The point on this curve which gave the minimum operating cost was then chosen as the proper thickness to use for the given conditions. The advantage of this method is that the curves are

[1] Research Engineer, Union Carbide and Carbon Research Laboratories, Long Island City, N. Y. Assoc-Mem. A.S.M.E.

[2] The Conservation of Heat Losses from Pipes and Boilers. Trans. A.S.M.E., vol. 40 (1918), p. 667.

not nearly so flat as those developed by the authors and it is much easier to determine the proper thickness to be used. The engineering profession has pretty generally recognized the enormous losses involved in the use of bare steam pipes, so that the present-day problem is more one of comparing the total operating cost with different kinds and thicknesses of coverings than of estimating the saving of the covered pipe as compared with a bare pipe. The difference in the two methods, however, is one of detail and the results should be the same whichever method is used.

The authors' paper shows the rate of heat transferred from bare iron pipe to air per linear foot of pipe in Fig. 2. It is important to note in this connection that while the heat loss per linear foot increases with the size of pipe due to the increased surface in the larger sizes, the rate of loss per unit of area varies in exactly the opposite direction; namely, the loss from small pipes is greater per unit of area than from large pipes. This makes the amount of heat which it is possible to save per square foot of small pipe greater than for a large one, but the cost of the covering per square foot in the smaller sizes is so much greater that unless the value of the heat units is very high the proper thickness of covering for maximum economy in the small pipe sizes will always be found to be less than that of the larger. The authors' conclusion as to the effect of the cost of heat on the most economical thickness of covering agrees in general with the results found at the Mellon Institute, but when the curves are plotted for the full range of pipe sizes from $\frac{3}{4}$ in. to flat surface and for the full range of temperature, it will be found in some conditions there is quite a decided variation in the most economical thickness. It should also be borne in mind that the trend of heat costs has been to increase for some time, and in general it would be well to put on a thickness of covering which is slightly greater than that indicated by the cost curve rather than to make it less than shown by the curve.

It is interesting to note that the authors decided upon 20 per cent of the cost of the covering as the annual fixed charges. Quite an extensive investigation was made on this phase of the problem at the Mellon Institute and the percentage finally decided upon was exactly the same.

The authors' demonstration that the drop in temperature from steam to the outside of the pipe does not affect the selection of the most economical thickness of pipe covering is interesting and the point is well taken, as this is a question about which there has been a great deal of misunderstanding. The conclusion which they have reached is undoubtedly correct.

This paper adds considerably to the information on pipe coverings and on heat flow in general, which has gradually been accumulated. It would be highly desirable if all this information could be brought together in a single paper, possibly in a book. There is a good opportunity available for someone to make this compilation.

RALPH J. HINCH.[1] The following comments are not strictly a discussion of the paper, but have a bearing on the subject of conductivity change of insulations over a long period of time.

About a year ago the Commonwealth Edison Company completed a series of tests, showing the conductivity of various pipe coverings. Each of the coverings was on test for about three weeks. The coverings were subjected to the maximum temperature of 750 deg. fahr. for only a small part of this time.

The question arose as to what influence the maximum temperature would have on the conductivity if allowed to run over a long period of time. Accordingly, four representative coverings were placed on test, and each was subjected to the 750 deg. fahr. temperature. To date the coverings have been on the test pipes for almost a year. During this period four tests have been made at intervals of two to three months. The second test showed over 7 per cent less conductivity than the original test in one of the coverings. The values for the other three checked the original very closely.

The third test showed a marked change in conductivity value for two of the coverings and no change in the other two. In each case of change, the conductivity value was increased, resulting in lower efficiency. The percentage differences over the second test were 6 per cent for the worse and 4 per cent for the other.

The fourth test showed practically the same values for conductivity as were obtained for the third test.

Summarizing, it can be said that, to date, two of the coverings showed practically no change in conductivity value. One showed a loss in efficiency when compared to the original. The last one showed a decrease in efficiency after the third test, which, however, was not so great as the gain in efficiency after the second test.

L. B. MCMILLAN.[2] The statements in Par. 23 regarding Fig. 6 are open to misinterpretation. The decision as to whether an additional thickness of insulation is justified cannot be based on the shape of the curve shown in Fig. 6. The general impression given by such a curve is largely dependent upon the scale chosen. Although the curve does flatten out very greatly as the thickness is increased, indicating that the saving by each additional increment of thickness is less than that by the preceding increment, this does not indicate that the additional increment is not justified unless the saving thereby is less than its cost per year.

As shown in Fig. 7, the curve of cost per year of various thicknesses of insulation shows a very low rate of increase also. Therefore, an increase in saving per year that might appear small when shown to the small scale used in the figure may actually repay

[1] Commonwealth Edison Company, Chicago, Ill.

[2] Consulting Engineer, Johns-Manville, Inc., New York, N. Y. Assoc. Mem. A.S.M.E.

considerably more than the cost per year. As the authors formed their conclusions upon this correct basis, their statements in Par. 23 should not be construed to mean that any definite conclusions might be drawn from Fig. 6.

R. J. S. PIGOTT.[1] There are several things in a paper of this character which it is well to take into consideration. Without question, a mathematical analysis is needed in such a subject as this, which is paralleled by many similar problems that have to be solved. For instance, the problem of how much circulating water should be used can be analyzed into a series of flat curves of the same character, the increase of power required for pumping causing an increase for operating cost, balanced against the saving in steam due to the increase of vacuum.

However, in addition we should consider several practical points regarding the insulation. First of all, no one cares to increase the fixed charge for the construction of the station, and where there is any great question about the saving it is considered wiser to keep on the low side rather than the high side of the investment cost. Therefore, the usual procedure would be to run to under-insulation rather than to over-insulation. Another point in favor of the lesser thickness of insulation is that it is more convenient and quicker to manufacture, and it is much quicker to repair.

Another question which affects the matter is that of where the heat losses are going. In many cases in the modern station, it is not a matter of very great importance, up to the point of discomfort to the operators, how great is the loss in the station, provided the heat lost goes back to the boiler. The only loss that cannot be recovered is that of radiation from the outside walls of the building. It will soon be common practice to see that all the ventilating air for the plant is recovered by the forced draft fans and returned to the boiler. In this case a slight change in loss from the piping is really not a net loss. It is preheating the furnace air and may result in a slight gain.

M. K. DREWRY.[2] That cognizance of losses other than pure heat losses should be taken into consideration when studying the economics of pipe covering is suggested in this discussion.

Suppose a loss of 25 deg. in superheated-steam temperature is experienced because of improper pipe covering. The resulting heat loss per pound of steam is approximately 13 B.t.u., or about 1 per cent of the heat added in the boiler room of the modern station. However, the turbine water rate increases approximately 2½ per cent as a result of the 25 deg. decrease in temperature,

[1] Consulting Mechanical Engineer, Public Service Production Company, Newark, N. J. Mem. A.S.M.E.

[2] Assistant Chief Engineer of Power Plants, The Milwaukee Electric Railway and Light Company, Milwaukee, Wis. Jun. A.S.M.E.

and the true energy loss is therefore 2½ times as large as given by the authors, warranting thicker coverings.

Superheaters are used to increase the availability of energy in the steam and to improve the efficiency with which turbines can transform that energy. Power-plant men are not interested in merely saving heat units, for in properly designed stations the pipe radiation losses may be recovered in the form of hotter combustion air. They are concerned, however, in obtaining the greatest electrical output with a given heat input, and consequently when designing pipe coverings a consideration of thermodynamic availability and turbine efficiency is necessary.

That pipe temperatures are within a few degrees of the steam temperatures within the pipe, as demonstrated by the authors, is very useful knowledge. To ascertain steam temperatures without having to remove headers from service for insertion of thermometer wells, and even to eliminate unreliable well installations for permanent service, the peening of thermocouple wires into the pipes is a desirable alternative.

E. T. Cope.[1] Mr. Hinch's remarks about the thermal conductivity brings out the fact that there is a great need for some research work to be done in some broad way to determine absolute thermal conductivity for these insulating materials. We should like to see, as users of these materials, agreement among the various people in attempting to determine absolute thermal conductivity.

Mr. Heilman has spoken of the futility of using a quarter of an inch of covering. There was no intention of advocating using a quarter of an inch. The intention was simply to point out that the curve of saving flattens off very rapidly, and that to put on an excessive thickness of covering is not advisable because it will not justify itself on an economic basis.

In connection with Mr. McMillan's remarks, Fig. 6 has to do with the heat saving and is not used otherwise except as a step in the solution. The most economical thickness of curve is shown in Fig. 7.

Mr. Pigott should note that labor cost ordinarily figures as a very large portion of the cost of installing covering, and in practically every covering put on a superheated steam pipe, there will be two layers of it at least. Whether the total thickness is 3½ in. or 4 in. makes no difference whatever in the labor cost. There is a slight difference in the matter of the cost of material itself, but not much. In regard to heating the air by losses from the steam piping, it should be noted that the air heating is being done by superheated steam, which it is desirable to get into the turbine at the maximum temperature. That steam is being heated in

[1] Authors' Closure.

boilers which operate at an efficiency of approximately 75 per cent. We consider that as a rather inefficient way to preheat air to be driven into these furnaces.

In answer to Mr. Drewry, let us calculate what would be the meaning of a loss of, say, 25 deg. fahr. in superheat from steam in a steam pipe. For the sake of a concrete example, let us consider a 16-in. steam pipe carrying steam at 720 deg. fahr. and passing 400,000 lb. of steam per hour.

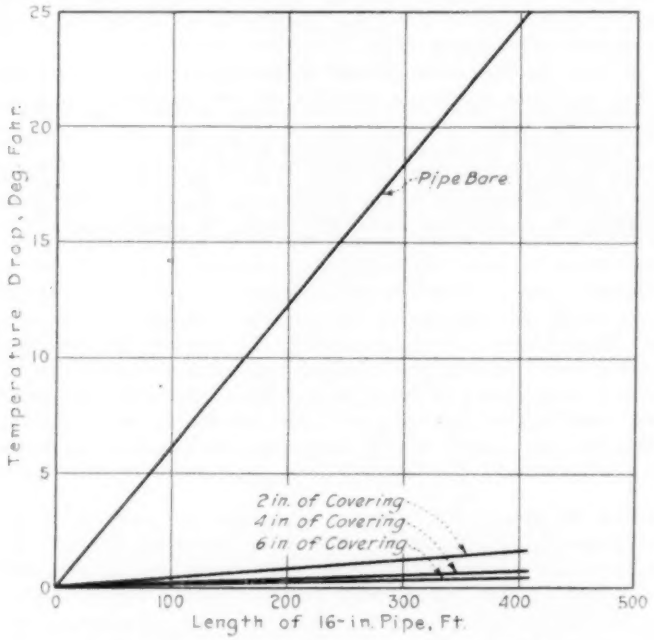

FIG. 13 TEMPERATURE DROP VS. LENGTH OF PIPE

16-in. pipe carrying 400,000 lb. of steam per hr. at 720 deg. fahr., pipe bare and covered.

Mr. Drewry's assumption of a loss of 25 deg. fahr. in superheat, or approximately 13 B.t.u. per pound of steam, anticipates a loss of $13 \times 400{,}000 = 5{,}600{,}000$ B.t.u. per hour through the pipe and covering.

In the study of steam-pipe covering for the Trenton Channel Plant, Heilman's constants for loss of heat from bare pipe were used. These were checked by Eberle's and Nusselt's formulas. The agreement was close. For a bare 16-in. pipe carrying steam at 720 deg. fahr. in air at 80 deg. fahr., the loss of heat was calculated to be 13,690 B.t.u. per running foot. The pipe temperature was taken the same as that of the steam.

In order that the loss of heat suggested by Mr. Drewry might occur, it would be necessary to pass the steam through a bare pipe, 409 feet long.

The same pipe, carrying steam as described at the rate mentioned and covered with 2 in. of a certain covering would lose 711 B.t.u. per running foot per hour. The temperature drop of the 400,000 lb. of steam per hour would be 1.65 deg. fahr.

If covered with 4 in. of covering as above, the loss per running foot would be 410 B.t.u. per hour. In the same length of pipe the temperature drop of the same 400,000 lb. of steam would be approximately 0.83 deg. fahr.

If now, the pipe were covered with 6 in. of this covering, the other conditions remaining the same, the temperature drop would be 0.62 deg. fahr.

These results are shown graphically in Fig. 13.

We must conclude that the effect of the drop of temperature on the steam consumption of the steam-using apparatus is negligible and that the question raised by Mr. Drewry is purely academic.

No. 1961

A REVIEW OF STEAM-TURBINE DEVELOPMENT

By Hans Dahlstrand,[1] Milwaukee, Wis.
Member of the Society.

The paper is chiefly concerned with the effects which the use of steam at higher pressures and temperatures has had upon the efficiencies of steam turbines and upon the materials used in the construction of their parts. The use of high-back-pressure steam turbines, to be installed in connection with boilers of higher pressure in such a way as to exhaust into the steam mains and turbines of existing plants, is carefully analyzed for units of 10,000- *and* 30,000-*kw. capacity, and the effects on the efficiency are graphically presented. A discussion of materials for use with the higher pressures and temperatures is included, with some curves showing the results of investigations made by the Allis-Chalmers Co. on various metals and the characteristics which should be possessed by a metal suitable for use at* 1000 *deg. fahr. The paper also contains some comments on corrosion and erosion of turbine blades, and shows the effects of these destructive forces on blades of different composition.*

THE constant increase in the use of electric energy for various purposes has forced an increasing demand for greater economy and a more reliable operation of all prime movers employed in producing this energy. Among these prime movers the steam turbine develops by far the largest amount of power used and on this account is receiving much attention from engineers engaged in the design of steam turbines and steam power plants. Keen interest is shown in all research, experiments, and new developments which tend to improve the economy and reliability.

2 While great improvements have been made, there are still great possibilities for further improvements, both in the design and in the application of steam turbines. This paper is an attempt to indicate a few of these possibilities, which may bring beneficial results, if some present difficulties can be successfully overcome.

[1] M. E., Steam Turbine Engrg. Dept., Allis-Chalmers Mfg. Co.

Contributed by the Power Division and presented at the Spring Meeting, Milwaukee, Wis., May 18 to 21, 1925, of The American Society of Mechanical Engineers.

PRESENT MAXIMUM STANDARD STEAM TEMPERATURE AND PRESSURE

3 It appears, from a general contact with work involving prime movers for the newer medium- and large-size power plants which have been proposed or constructed during the past two to four years, that the present normal maximum steam temperature which may be regarded as commercially practicable is about 700 deg. fahr., with temperatures ranging from 725 to 750 deg. fahr. for limited periods, and that the pressure is and has been in more of a stage of transition. Considering installations as a whole, the maximum standard pressure is generally about 350 lb. gage at the the turbine, with many installations contemplating pressures of 385 lb. gage.

4 Turbines for these steam pressures and temperatures have been in successful operation in many plants; and for sizes up to

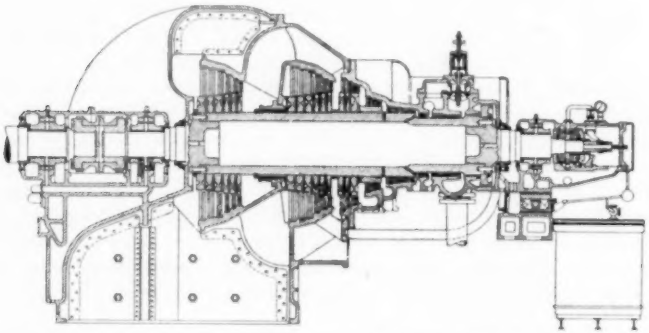

FIG. 1 30,000-KW., 1800-R.P.M. STEAM TURBINE, 350 LB. GAGE PRESSURE, 240 DEG. FAHR. SUPERHEAT, 29 IN. VACUUM

50,000 kw. it appears that the turbine for 350 lb. gage and 700 deg. fahr. will, in nearly all cases, be of the single-cylinder, single-exhaust-nozzle type. Fig. 1 shows a section through such a machine having a single cylinder and a single and unidirectional double-flow steam arrangement. The general efficiency characteristics and expansion through such a turbine is shown as curve A on the Mollier diagram, Fig. 2.

5 Developments during the past several years have indicated the feasibility of employing steam pressures of 500 to 600 lb. gage, with temperatures in the 700-deg. fahr. field. In this country two new plants of these general characteristics have been put into successful operation. These plants are designed to accomplish eventually reheating at a pressure of 120 lb. gage after expansion from 550 lb. gage, the steam temperature at this pressure being raised again to 700 deg. fahr., and the expansion completed to a vacuum slightly greater than 29 in.

6 The turbine arrangement for this type of plant has a greater divergence than is the case for the 350-lb. units. For instance, the two 550-lb. plants in this country contain single-cylinder, two-cylinder, and three-cylinder turbines, with both single and double flow of the steam. The general characteristics of efficiency and expansion for a turbine operating on the reheat cycle outlined in the foregoing is shown as curve B on the Mollier diagram,

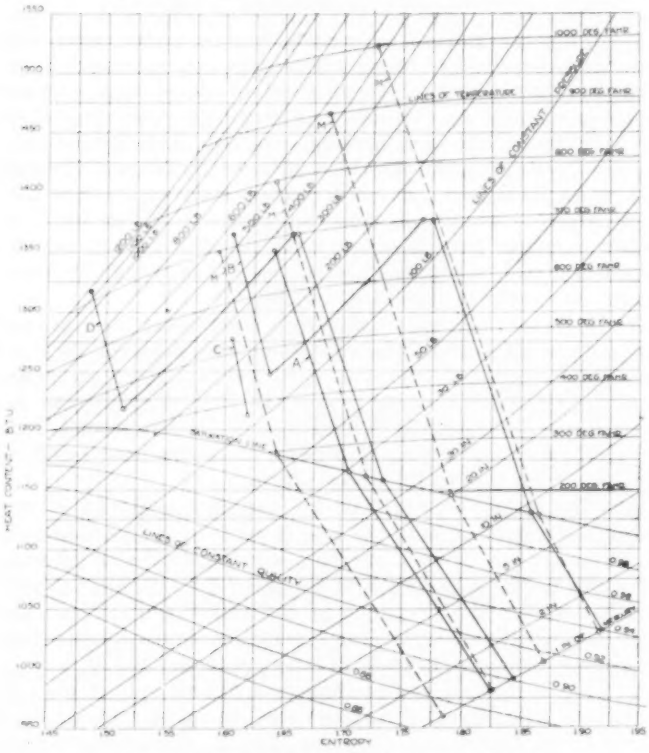

FIG. 2 VARIOUS TYPES OF EXPANSION LINES

Fig. 2. This particular expansion line was drawn from estimates made for a two-cylinder, two-generator, cross-compound 50,000-kw. unit.

HIGHER-STEAM-PRESSURE INSTALLATIONS

7 Development work looking to the use of still higher pressures and temperatures is constantly in progress with general indications that immediate steps will be taken in the direction of increasing the pressure at about the present maximum temperature. The author can add nothing to the very complete descrip-

tions which have been published of the highly interesting installation under construction at the Weymouth power station of the Boston Edison Company, with a 1200-lb. boiler and a high-back-pressure turbine. By the time this paper is presented this installation may be in operation, and its operation will be a matter of the keenest interest to all engineers.

High-Back-Pressure Steam Turbines

8 It has always been appreciated that in plants using large quantities of steam for heating and process work at comparatively high pressures it would be economical to install steam turbines receiving steam from the boilers at very high pressures and exhausting against the back pressure required. The calculation of the gain to be obtained from such a turbine installation is obviously a very simple matter, but until recently such applications have been largely prevented because no standard boilers for these higher pressures have been available. In addition the engineer considering such a problem has not been able readily to obtain figures on a suitable steam-turbine unit.

9 However, there has been at least one pioneer application of this type of apparatus. The rubber industry centered at Akron uses large quantities of steam at about 100 lb. pressure in connection with the manufacturing process. In spite of a relatively low initial boiler pressure, 250 lb. gage, one of the Akron mills found it feasible to install a turbine between the boiler header and the pipe supplying the 100-lb. process steam.

10 The turbine was of 5000 kw. capacity, operating at 1800 r.p.m. and regulated as to the load it carried by a back-pressure regulator, as well as by a standard centrifugal type of governor. This unit has been in successful operation for a number of years and has confirmed the opinion of the author and his associates that the highest turbine efficiency would be obtained in the high-pressure sections, or elements, of large steam turbines.

11 Fig. 3 shows a section through a turbine unit of the type described. The efficiency characteristics of the unit are shown as curve C on the Mollier diagram, Fig. 2. Observations made during regular operation show that the turbine efficiency ranges between 75 and 85 per cent, the observations being based on both measurements of the quantity of steam and on readings of inlet- and exhaust-steam thermometers, the turbine exhausting superheated steam.

12 While the unit described has not operated with the higher pressures and temperatures now current, a design of the same characteristics will not have its efficiency impaired by the higher pressures, nor the greater density of the steam, since proportionately the surfaces over which the steam flows and which are responsible for the losses will be reduced. Further, with a proper

design the steam velocity will be somewhat lower at the higher pressures.

High-Back-Pressure Turbines in Existing Power Plants

13 In view of the successful application of high-back-pressure turbines the author has felt that it would be worth while considering the possibilities in the suggestion that has been made of applying high-pressure units in the remodeling of existing plants. Many older plants with moderate steam pressures and temperatures have been equipped with steam turbines of designs which have shown, by test, efficiencies that will be little improved upon, at least in the near future. Such installations are operating with condensing equipment which, in so far as the vacuum produced is concerned, will not be materially improved upon.

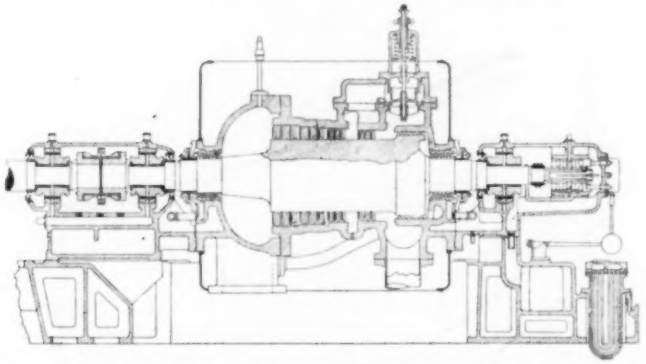

Fig. 3 5000-Kw., 1800-R.p.m. Steam Turbine, 250 Lb. Gage Steam Pressure, 125 Deg. Fahr. Superheat, 100 Lb. Gage Back Pressure

14 Without having made an analysis of the cost considerations outside of the turbine equipment, it appears that the problem would be worthy of detail solution if a high-efficiency, lower-pressure, existing plant were considered as being obsolete only in respect to the boiler-room pressure and temperature. The remodeling would then consist of the installation of new boiler-room equipment operating at a higher pressure and temperature, and a high-back-pressure turbine taking steam from the boilers and exhausting into the older and lower-pressure header system.

15 With the hope that the problem outlined might be taken up and analyzed, the author has made certain efficiency comparisons which may be of service in considering an estimate for remodeling older plants. For the general cases considered, the new boiler pressure, which has been called the primary pressure, might be 550 lb. gage or greater, and the steam temperature about 700 deg. fahr. The proposed high-back-pressure element

would be designed to exhaust at the original boiler pressure. No account has been taken of the gain to be obtained by the possibility of using extracted steam for feedwater heating for the older or existing units. Considering this feature alone, the estimates as to the gains in economy will be found very conservative.

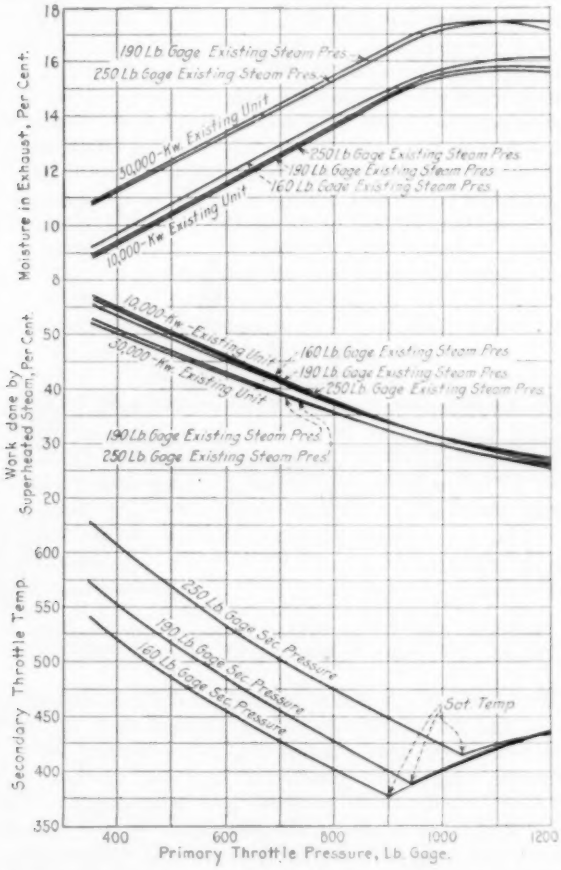
Moisture in Exhaust, Per Cent.
Work done by Superheated Steam, Per Cent.
Secondary Throttle Temp.
Primary Throttle Pressure, Lb. Gage.
190 Lb Gage Existing Steam Pres.
250 Lb Gage Existing Steam Pres.
30,000-Kw. Existing Unit
10,000-Kw. Existing Unit
250 Lb Gage Existing Steam Pres.
190 Lb Gage Existing Steam Pres.
160 Lb Gage Existing Steam Pres.
160 Lb Gage Existing Steam Pres.
190 Lb Gage Existing Steam Pres.
250 Lb Gage Existing Steam Pres.
190 Lb Gage Existing Steam Pres.
250 Lb Gage Existing Steam Pres.
250 Lb Gage Sat. Pressure
190 Lb Gage Sat. Pressure
160 Lb Gage Sat. Pressure
Sat. Temp

FIG. 4 CHARACTERISTIC STEAM CONDITIONS OF EXISTING STEAM PRESSURES WITH VARIATIONS IN PRIMARY THROTTLE PRESSURES AT CONSTANT TOTAL STEAM TEMPERATURE OF 700 DEG. FAHR.

16 In general the analysis has been made to cover two existing sizes of turbines, 10,000 and 30,000 kw., and the information given shows the effect of the efficiency of such existing turbines on the gain to be expected if a high-pressure remodeling were put into effect.

17 The efficiency of steam turbines is affected to the greatest extent by the percentage of the total work of the turbine which is done by superheated steam. The greater the percentage of work done by superheated steam, the less will be the mean percentage of moisture in that part of the turbine using saturated steam. Fig. 4 shows the effect of primary throttle pressure on the percentage of work done by superheated steam, the moisture in the exhaust

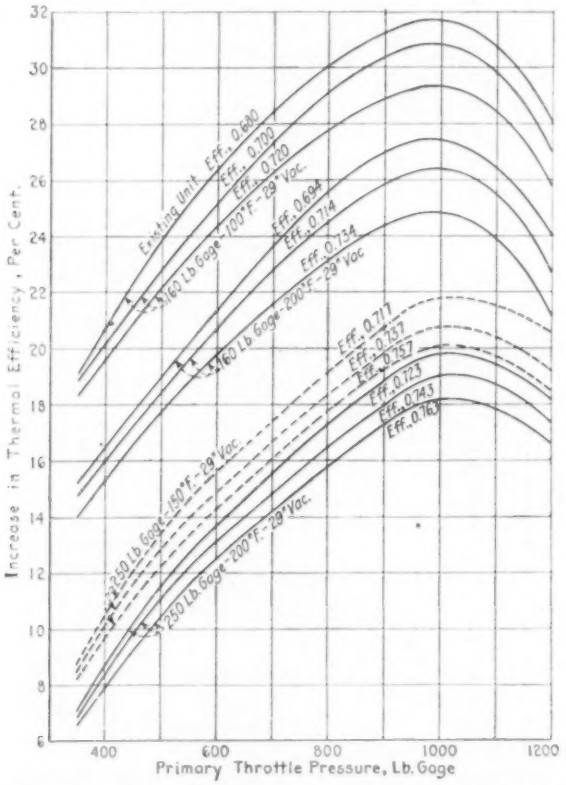

FIG. 5 INCREASE IN THE EFFICIENCY OF THE COMBINED UNIT OVER THE EXISTING UNIT AT PRESENT STEAM PRESSURES AND TEMPERATURES, BASED ON VARIOUS PRIMARY THROTTLE PRESSURES AND TOTAL STEAM TEMPERATURE OF 700 DEG. FAHR.

steam, and the temperature of the steam exhausted from a high-back-pressure steam-turbine element for the two different sizes of turbine units mentioned and for several secondary or existing steam pressures. It will be noted that for a primary throttle pressure of 550 lb. gage the secondary throttle temperature and the moisture in the exhaust steam cover the normal range for the

older high-efficiency plants; that is, the secondary throttle temperature from 475 deg. at 160 lb. gage to 550 deg. at 250 lb. gage. The analysis covers 190 and 250 lb. gage steam pressures for

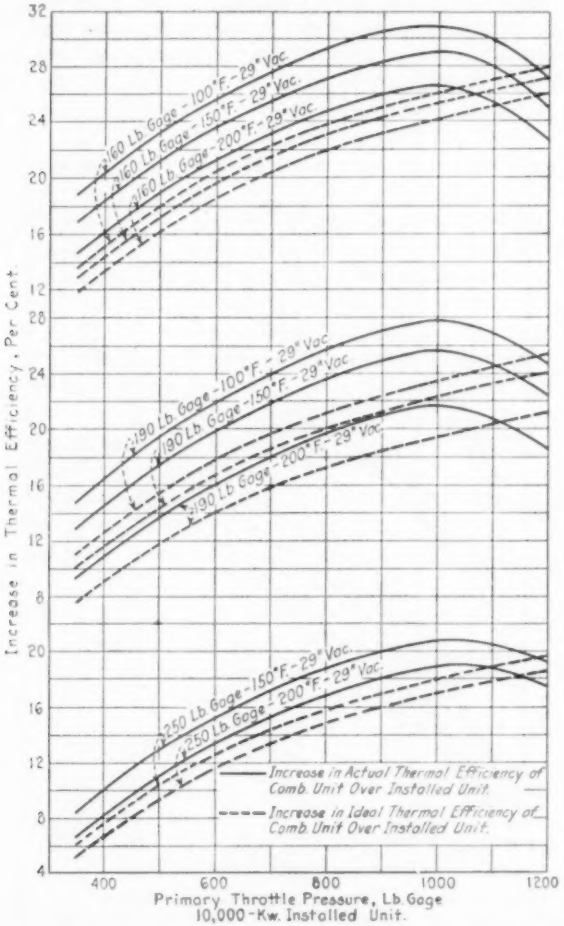

FIG. 6 INCREASE IN THE EFFICIENCY OF THE COMBINED UNIT OVER THE EXISTING UNIT AT PRESENT STEAM PRESSURES AND TEMPERATURES, BASED UPON VARIOUS PRIMARY THROTTLE PRESSURES AND TOTAL STEAM TEMPERATURE OF 700 DEG. FAHR.

10,000-kw. and 30,000-kw. units with an additional steam pressure of 160 lb. gage for 10,000-kw. units.

18 Fig. 4 shows generally that for the highest pressure for which boilers are commercially available, 550 lb. gage, the re-

modeling of a boiler room and the installation of a high-back-pressure turbine would introduce no operating conditions which would cause an average existing turbine unit to suffer a loss in economy.

19 Taking full account of those factors which influence turbine efficiency, calculations have been made to show the increase in

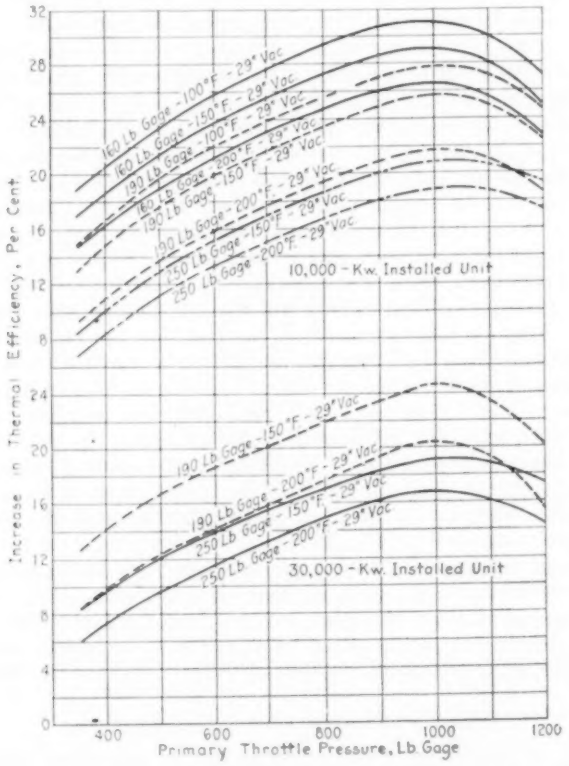
Increase in Thermal Efficiency, Per Cent
Primary Throttle Pressure, Lb Gage
160 Lb Gage - 100°F - 29" Vac
160 Lb Gage - 150°F - 29" Vac
160 Lb Gage - 100°F - 29" Vac
160 Lb Gage - 200°F - 29" Vac
160 Lb Gage - 150°F - 29" Vac
190 Lb Gage - 200°F - 29" Vac
250 Lb Gage - 150°F - 29" Vac
250 Lb Gage - 200°F - 29" Vac
10,000 - Kw. Installed Unit
190 Lb Gage - 150°F - 29" Vac
190 Lb Gage - 200°F - 29" Vac
250 Lb Gage - 150°F - 29" Vac
250 Lb Gage - 200°F - 29" Vac
30,000 - Kw. Installed Unit

FIG. 7 INCREASE IN THE EFFICIENCY OF THE COMBINED UNIT OVER THE EXISTING UNIT AT PRESENT STEAM PRESSURES AND TEMPERATURES WITH VARIATION IN THERMAL EFFICIENCY OF EXISTING UNIT BASED ON VARIOUS PRIMARY THROTTLE PRESSURES AND TOTAL STEAM TEMPERATURE OF 700 DEG. FAHR.

thermal efficiency which might be expected if higher boiler pressures and high-back-pressure turbines were applied to existing plants, using units of 10,000- and 30,000-kw. capacity. The range of pressures has extended to 1200 lb. gage, the primary initial steam temperature being, for all pressures, about 700 deg. fahr.

The calculations have been based on expansion in the high-back-pressure turbine to different exhaust pressures ranging from 160 to 250 lb. gage. The results of these calculations are summarized graphically in Fig. 5. It is interesting to note the falling off in the increase in thermal efficiency which occurs in the higher pressure range above 1000 lb. gage. In general it may be said that this decrease in the gain to be expected results from the greater proportion of the work which is done by moist steam when the pressure is increased and resuperheating is not resorted to.

20 The increase in thermal efficiency for different primary throttle pressures as approximately estimated for actual turbine installations checks in general characteristics with that indicated by theory for prime movers of 100 per cent efficiency. As a matter of comparison, the probable actual gain and the theoretical gain for the use of higher steam pressures and high-back-pressure turbines with an existing plant have been summarized graphically in Fig. 6. It will be noted from this diagram that the actual gain is, for the most part, greater than the theoretical, a condition which is obvious when it is considered that the inefficiency of the lower-pressure existing machine will increase the gain to be made.

21 Mention has been made of the fact that the inefficiency of an existing plant will increase the gain to be expected over that indicated by theory if the existing turbine is combined with a high-back-pressure turbine as a unit. The estimates and calculations have been made for 10,000- and 30,000-kw. units, the chief difference between these being a difference in the assumed efficiency. Calculations showing the effect of the lower-pressure or existing turbine's efficiency on the gain to be expected from the higher-pressure combination under discussion have been made for a limited range of turbine efficiency, 4 per cent, and for steam pressures from 160 to 250 lb. gage at different superheats. The results of these calculations are shown in Fig. 7, and the author feels that such curves may be found of some service to engineers engaged in investigating different methods of improving plant efficiency.

Higher-Temperature Development

22 In considering a few of the details in connection with the improving of plant efficiency, the author has confined his remarks largely to the possibilities of increasing the pressure, and, to a more limited extent, of increasing the temperature to a normal maximum of about 700 deg. fahr.

23 It has been mentioned that the percentage of work done by superheated steam for a given condition has a marked effect on the average moisture in the steam in the lower-pressure stages of a turbine. As is well known, if the steam temperature is increased at constant pressure there will result a marked decrease in the moisture found in the steam exhausted from the turbine. It has

also been mentioned as a well-known theory, which tests have demonstrated to be also an established fact, that the efficiency of the turbine is greatly affected and benefited by a reduction in the amount of work done by moist steam. It follows, therefore, that the theoretical increase in thermal efficiency which occurs for an increase in initial temperature will be exceeded in the performance of an actual machine.

24 Taking the effect of moisture on the stages of a turbine at a value which has been confirmed by tests, the increase in thermal efficiency which will most probably be obtained if steam at a pressure of 550 lb. gage is increased in temperature from 700 to 1000 deg. fahr. is shown by the upper curve of Fig. 8. The lower dotted line of Fig. 8 shows the theoretical increase in thermal efficiency.

25 The form of the expansion lines and the efficiency characteristics for the different turbines considered, including those

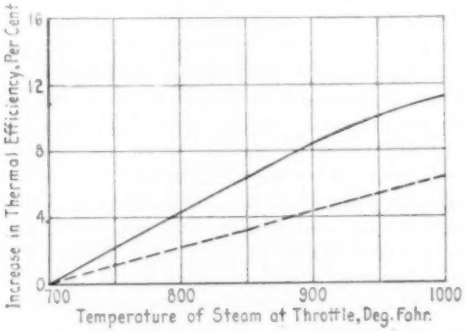

FIG. 8 INCREASE IN THERMAL EFFICIENCY AT CONSTANT PRESSURE WITH INCREASING TEMPERATURE, BASED ON 550 LB. GAGE AND 700 DEG. FAHR.

operating at proposed temperatures of 1000 deg. fahr., are all designated by the letter *M* on the Mollier diagram, Fig. 2.

26 The figures given for probable turbine efficiencies in excess of the 700-deg. range are all strictly tentative in that the author is not in possession of sufficient information regarding materials to warrant his stating that the construction of a 1000-deg. turbine at present would be justified on any basis other than experimental development.

27 Curve D of Fig. 2 shows an expansion line worked up from estimates made for a 1200-lb., high-back-pressure, proposed turbine installation, and it is felt that it represents fairly well the efficiency that may be expected with a machine designed for these conditions and not to function primarily as a reducing valve. The gain in thermal efficiency as indicated on the curves refers to turbine alone and does not include boilers or auxiliaries.

Effect of Moisture in Saturated Steam

28 The known effect of moisture as a factor which limits the attainable efficiency of a steam turbine has directed considerable attention to the gain that would result if it could be eliminated or its quantity reduced in the lower-pressure, high-velocity stages. For turbines operating in plants having superheated steam the moisture is formed as the result of work being done. For a given superheat the amount of moisture will depend on the efficiency of the turbine: the greater the efficiency, the greater the moisture at exhaust.

29 From these considerations it appears that the best remedy that can be hoped for in a given installation would be a device or construction which would remove the moisture at the exhaust of one stage so that it will not be added to that which will be formed in the stages following. Such an arrangement would probably result in a gain, because it is also known that the loss in efficiency in a given stage increases with the percentage of moisture.

30 Judging from his own experience and from a study of the experience of others in the separation of moisture from steam, the author believes that the present construction of turbines does not offer those characteristics which have been essential in other apparatus to the successful separation and removal of moisture. The most effective separating devices have uniformly depended on a low velocity of the steam and, in some cases, the employment of large surfaces in combination with low velocity. These two features are not practicable in the present designs in which commercial considerations make it essential to put the maximum possible number of low-pressure stages in a single cylinder.

31 It is true that the flow of steam through the lower-pressure stages concentrates the moisture near the periphery of the turbine blading, but the same observations which have shown this concentration to be a fact have also shown that the cylinder wall of the turbine is not covered with a continuous film of water. In accordance with these views the author believes that different methods of extracting steam at interstage points will have but a slight effect on the efficiency of the low-pressure stages of a given turbine. Further, there seems to be no immediate prospect of a design permitting of the application of effective moisture separating and removing features becoming commercially feasible in the near future.

Auxiliary Turbines for Power-Plant Service

32 The author has found through contact with power-plant engineers that the subject of a reliable and economical drive for the different essential continuous-service auxiliary apparatus is still in the controversial stage, and that opinions differ greatly, with some indication that the small turbine for this service may again find favor. The use of the electric motor for auxiliary drive

has no doubt been extended through the rapid development of feedwater heating by interstage extraction of steam from the main turbine and to a lesser extent by the use of direct-driven auxiliary generators. These two arrangements not only increased the application of the electric motor to auxiliary service but tended to supplant the steam-driven house generator set as a source of power for the electrically driven auxiliaries.

33 There is no question but that the smaller-size turbine for direct connection to the larger and essential auxiliaries can be

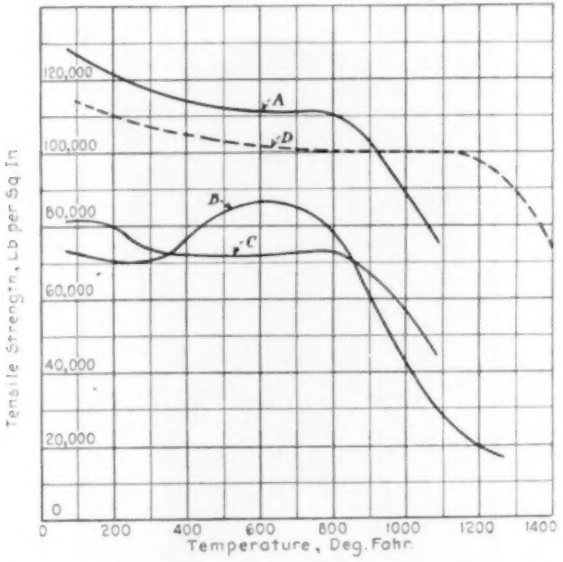

FIG. 9 PROPERTIES OF METALS AT HIGH TEMPERATURES

A Rolled High-Chrome, High-Nickel Steel
B Forged 3½ Per Cent Nickel Steel
C Forged Monel Metal
D Metal Suitable for Higher Temperatures

vastly improved as to both economy and reliability if there proves to be a real demand for it. The turbine designer, if given the opportunity, can meet such a demand with a small-power turbine which will be equally or more reliable than the main unit which it is serving. Further, this small turbine can be designed with an efficiency which will make it a satisfactory apparatus for a modern plant.

34 So far as the author knows, the power-plant engineer has not been sufficiently committed to the use of the small steam turbine in large modern plants to warrant the work which is necessary for the development of even tentative detail designs, although there are some indications that it will be practical to

undertake such work in the near future. To take full advantage of the possibilities in small-turbine design and its application to the driving of power-house auxiliaries, it will be essential to make full use of reduction gearing of proportions which have proved to be good for continuous service.

Special Steam-Turbine Materials

35 The use of cast steel in place of cast iron for the high-pressure, high-temperature sections of a steam-turbine cylinder

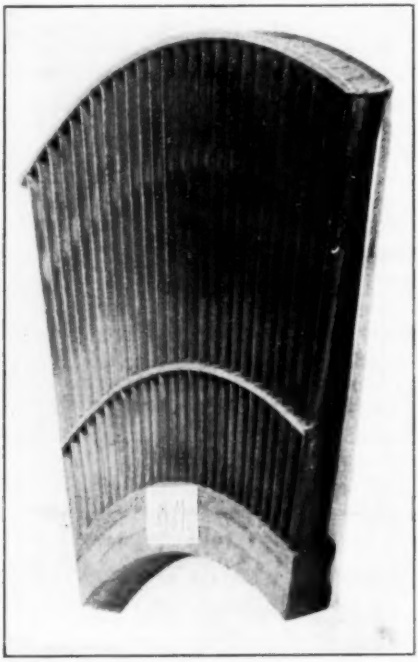

Fig. 10 Effect of Erosion on Turbine Blading
(Monel Metal)

began with the advent several years ago of superheats producing steam temperatures of 500 deg. fahr. or more. Cast steel is still employed for the cylinder or stationary parts for the highest temperatures now used, but in the author's experience it has been found of great value to heat-treat these cast-steel parts for 700-deg. service in a manner which results in a structure of the material itself that is more stable for long service. It has further developed that the expense of annealing the cast-iron parts of the major sizes of turbines is warranted, and adds to the assurance that the turbine, as a whole, will be made up of parts whose shapes

and dimensions are stable. The research engineers of the company with which the author is associated found that the heat-treating of cast-steel parts, and the annealing, in a proper manner, of cast-iron parts called for investigations not covered by published information. The work on the annealing of cast iron was of especial interest and has been published separately.[1]

36 The research department of this company has also carried out extensive tests on the effect of temperature on the strength and other properties of different materials. The results of these

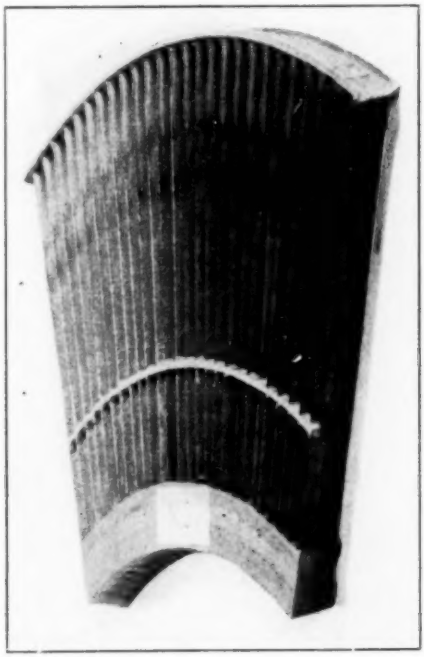

FIG. 11 EFFECT OF EROSION ON TURBINE BLADING
(3½ Per Cent Nickel Steel)

tests have been compared as to accuracy with those carried out independently by other investigators. The strengths of three materials, rolled high-chrome, high-nickel steel, forged 3½ per cent nickel steel, and forged monel metal at different temperatures are shown graphically in Fig. 9, on which there are also indicated as curve D the properties, so far as tensile strength is concerned,

[1] Annealing Gray Cast Iron, by J. F. Harper and R. S. MacPherran, presented before the American Foundrymen's Association meeting, Rochester, N. Y., June 5-9, 1922. Reprinted in Allis-Chalmers Bulletin 141, Research in Materials, Nov., 1922.

which are considered desirable for a material suitable for steam-turbine service with a temperature of 1000 deg. fahr.

37 There are in progress additional tests in which is being determined the behavior of different materials when subjected to sustained or continuous stresses at different constant and high temperatures. These tests also cover the effect of repeated heating and cooling by steam of steam-turbine structures. The author regrets that, since such tests must necessarily cover a very extended

FIG. 12 EFFECT OF EROSION ON TURBINE BLADING
(80 Per Cent Copper, 20 Per Cent Nickel)

period, it is not possible at this time to present the facts which these tests are expected to establish.

38 The development of satisfactory turbine structures has also been found to depend upon the resistance of the different materials to repeated or reversed stresses and elaborate tests have been in progress over several years to determine the resistance of the materials employed to fatigue or progressive failure. Fig. 14 shows plainly the development which has been made in this respect, the lower curve, C, being for a copper-nickel alloy which was extensively used for many years and the upper curve, A, being for an

excellent high-chrome, high-nickel steel developed for steam-turbine-blading service.

Development of Non-Corrosive Materials

39 The steam-turbine field has apparently been looked on by those engaged in the development of non-corrosive metals as the promising market and their products have found application as designs have progressed and required materials having greater

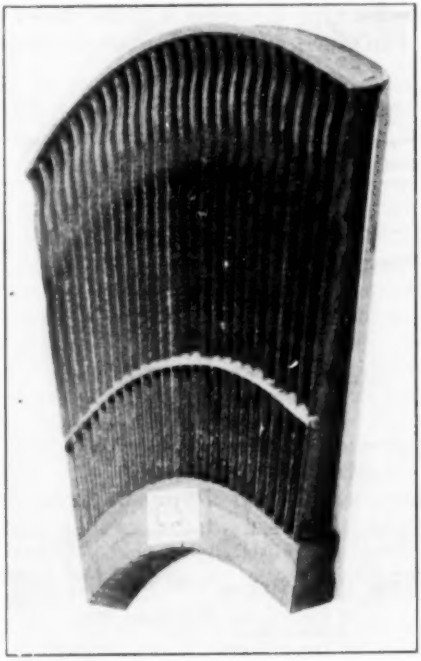

Fig. 13 Effect of Erosion on Turbine Blading
(Ordinary Carbon Steel)

strength, as well as resistance to corrosion, than the copper and nickel alloys which have been in such long and successful use.

40 In considering the application of the newer non-corrosive ferrous alloys to steam-turbine work, the author feels that there has been considerable confusion between corrosion and erosion. Non-corrosive steels have, in general, found their chief application for turbine blades and buckets and for valve-trim material. A careful review of the records of machines long in service indicates that the blading material or bucket construction of a turbine is more often and most severely attacked by erosion with corrosion

as a secondary action. These remarks are to be taken as applying to machines which have not been equipped with ordinary carbon-steel blading or buckets, since all experience makes it obvious that thin-edged blades and buckets made of a steel which is subject to the slightest corrosive action would prove to have, in most cases, a relatively short life. The turbine industry from its foundation recognized the importance of non-corrosive materials for the main construction, and the author feels that all departures from this fundamental principle must have been matters of expediency.

41 The reaction type of turbine, with the design of which the author has had his experience, has the erosive action on the turbine

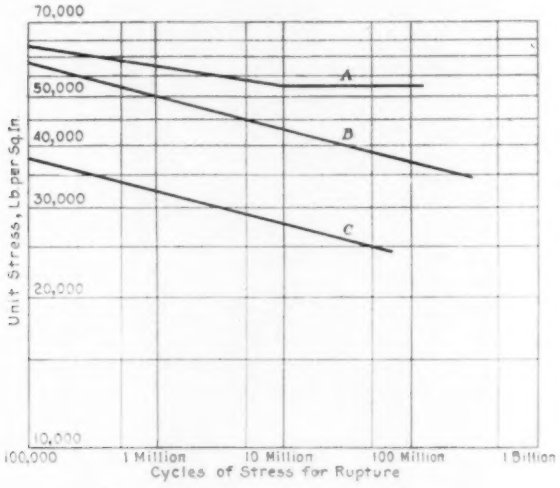

FIG. 14 EFFECT OF REPEATED STRESSES

A High-Chrome, High-Nickel Steel
B Monel Metal
C Copper-Nickel

blading inherently confined to a very few stages at the exhaust end of the machine. This condition has led to the carrying out of corrosion tests on finished turbine blading material in service under the most severe conditions. These tests have been limited in scope because of the limited range in the reaction type of turbine in which the erosive action manifests itself on even the softest material. Corrosion, in the author's experience, has always been taken care of by the employment of materials which are not affected by the active corrosive agents ordinarily found in steam supplied a turbine.

42 The service tests mentioned have consisted of installing in the last three stages of a 3200-kw. turbine, alternate blade

sections of different materials. This turbine uses a blade having a length of 6 in. for the stages subjected to the tests. The service is unusually severe since the machine operates condensing six months of the year and non-condensing the remaining six months. The results of one such series of tests are apparent from the photographs reproduced. Fig. 10 shows a section of monel-metal blading, Fig. 11, one of 3½ per cent nickel steel; Fig. 12 one of an alloy, 80 per cent copper and 20 per cent nickel; and Fig. 13, ordinary carbon steel. The materials and figure numbers are mentioned in the approximate order of their resistance to erosion as manifested by a wearing away of the inlet blade edge directly under the shroud ring at the blade tip. The nickel steel and carbon steel, Figs. 11 and 13, were also subject to considerable corrosion, the results of which do not show plainly on the photograph, but which was sufficient to make them unsatisfactory as compared with non-corrosive material for turbine-blade construction.

43 All of the materials subjected to the service test just described were installed in the same machine adjacent to each other and were operated in the condensing and non-condensing service which has been mentioned. Additional service tests are in progress and these, in the author's opinion, form the only satisfactory method of determining the advantages and disadvantages of different materials for special blade service.

44 It is well known that cast iron maintains its tensile strength up to a comparatively high temperature, and from this fact it appears that this material would be suitable for a considerably higher temperature; but cast iron will gradually change, increasing in volume even when exposed to comparatively low temperatures over long periods of time, and is therefore not suitable except for low temperatures. It is probable that other materials now considered suitable for high temperatures, or new materials now being developed, will have similar characteristics when exposed to high temperatures over long periods of time, and it is therefore of considerable importance that this characteristic of the material be thoroughly investigated. Undoubtedly the heat-treated materials must be tested to see if, at comparatively low temperatures, the physical characteristics will gradually change. It is obvious, therefore, that a great deal of research and investigation are needed before we can, with full confidence, go ahead with the construction of machinery for power plants using temperatures appreciably higher than those now being used.

45 In conclusion the author wishes to express his appreciation to Prof. A. G. Christie, Mr. E. H. Brown, and others, for valuable suggestions and assistance in preparing this paper.

DISCUSSION

A. G. CHRISTIE.[1] The author has presented in this paper many interesting data on modern steam turbines, most of which need no comment. He presents in Par. 11 data on the turbine efficiency of the high-back-pressure turbine at Akron which range from 75 to 85 per cent. These figures will assist in dispelling the idea still held by some engineers that the high-pressure sections of a turbine are not efficient.

The author's proposal to install high-back-pressure turbines in existing plants is an interesting one and his curves merit careful study. He is apparently satisfied that this would pay as far as the turbines are concerned. However, this proposal involves a reconstruction of the boiler plant for the new high-pressure boilers. If an engineer were given such an opportunity to reconstruct his boiler plant along modern lines, he would probably want larger boilers with higher settings than in his old plant. It probably would not be possible to make such changes with his present column centers, roof height, coal bunker, and coal-handling equipment. Reconstruction is an expensive item, and it is doubtful whether the complete costs of changing an old plant to high pressure would warrant such an outlay. When an expenditure of such an amount is to be made, it would seem more advisable, in view of the present state of electrical power generation, to hold the old low-pressure station for stand-by peak-load service and to design and build a complete new high-pressure section or station that could be used on base-load service. The engineer would then have complete freedom in selecting the most suitable boilers, furnaces, pumps, etc.

The author's comments on higher steam temperatures are of interest. The data shown in Fig. 8 refer to the expansion curves shown in Fig. 2 for various high temperatures at 550 lb. absolute pressure. These data in Fig. 8 apparently refer to turbine performance only, and to expansion on the Rankine cycle. The writer has had calculations made on the probable actual gains in efficiency for the whole plant with 400 lb. absolute pressure and temperatures up to 1000 deg. fahr. on both Rankine and regenerative cycles. The increases in thermal efficiency from these total plant calculations are shown in Fig. 15, together with a dotted curve reproduced from Fig. 8, giving the author's results. A comparison of these curves indicates that the increase in efficiency with rising temperature for the whole plant at 400 lb. on the Rankine cycle is about 1 per cent less than in Fig. 8 until 1000 deg. fahr. is reached, when the two curves converge. The increases in efficiency with rising temperatures on the regenerative cycle are from

[1] Professor of Mechanical Engineering, Johns Hopkins University, Baltimore, Md. Mem. A.S.M.E.

1.5 to 2.5 per cent less than those shown by the author in Fig. 8. All these data, however, show substantial gains from increased superheat and should encourage attempts to raise our present accepted temperature limits. Furthermore, increased temperature will result in materially decreasing steam consumption. Smaller condensers will be required and less cooling water per kilowatt capacity. Where cooling water is limited, increased steam temperature will permit increased plant output with a given amount of cooling water. These considerations will be leading factors in many power-station designs in the near future.

The author's comments on the effect of moisture in saturated steam in Pars. 28 to 31 would indicate that he has no hope of separating moisture from exhaust steam in a turbine. Some British designers have provided devices to effect at least a partial separation of moisture and to drain it to the condenser. It would

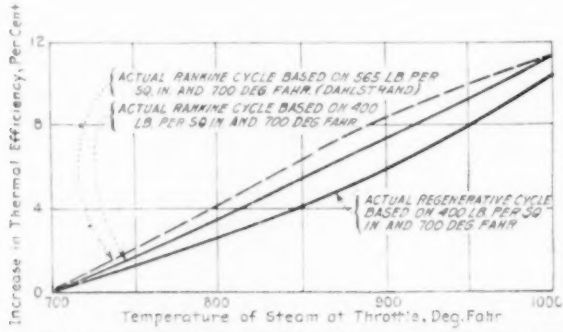
Increase in Thermal Efficiency, Per Cent
12
8
4
0
ACTUAL RANKINE CYCLE BASED ON 565 LB. PER SQ. IN. AND 700 DEG. FAHR. (DAM. ST. HAND.)
ACTUAL RANKINE CYCLE BASED ON 400 LB. PER SQ. IN. AND 700 DEG. FAHR.
ACTUAL REGENERATIVE CYCLE BASED ON 400 LB. PER SQ. IN. AND 700 DEG. FAHR.
700
800
900
1000
Temperature of Steam at Throttle, Deg. Fahr.

FIG. 15 COMPARISON OF THERMAL EFFICIENCIES OF VARIOUS CYCLES AT INCREASING STEAM TEMPERATURES

be interesting to learn whether these have had any effect on efficiency. Since the moisture in the low-pressure stages of the turbine is concentrated near the periphery, some slight gain may be secured by bleeding such wet steam to heat feedwater.

With regard to auxiliary turbines for power-plant service, there are still places in a power plant where a reliable, highly efficient, auxiliary turbine could be used. It does not seem good engineering to install highly efficient plant equipment throughout and then add an inefficient auxiliary turbine which will be in constant use. There seems to be a real field for more efficient small turbines than those now on the market, not only in central stations, but also in marine service.

Much research work must still be done on steam-turbine materials, and the work recorded in this paper is a valuable contribution to the subject. Increased reliability results from the use of better and more suitable materials. New data are particularly needed

on the action of various metals under higher temperatures, for the use of such steam temperatures is wholly dependent on finding metals that will provide a reliable turbine construction. In particular, information is needed on the effect of constant stress at high temperature when the so-called "creep" phenomenon occurs, and also on the effect of repeated or alternating stress on materials at high temperature.

E. H. THOMPSON.[1] One fundamental point stands out in the author's excellent and helpful analysis of conditions leading up to the application of higher pressures and temperatures. Something must be done other than increase pressures if overall efficiencies are to be improved beyond those now commercially practicable at 700 deg. fahr. and about 1000-lb. per sq. in pressure. Reheating makes possible a practical gain, due to reduction of condensation in the lower stages, but the theoretical gain is negligible. Increasing the initial temperature, however, is more advantageous with regard to cyclic efficiency, but this advantage is not marked at the higher pressures considered, as compared to the lower.

Figs. 5, 6, and 7, which show in various ways the advantages of increased pressures at 700 deg. fahr., all indicate a falling off above 1000-lb. per sq. in. pressure. The gains characteristic of station practice drop more sharply than those of theoretical conditions, the result, as the author points out, of friction in the lower pressure stages owing to water. It would appear that at not far above 1200-lb. per sq. in. pressure the region of zero gain would be reached.

The secondary throttle would here receive moist steam as indicated in Fig. 4, reheating not being considered.

The author indicates the beneficial effect of temperatures above 700 deg. fahr. at a pressure of 550 lb. per sq. in. in Fig. 8. The gain due to the reduction of friction in the lower-pressure region is roughly equal to the theoretical gain. In connection with this part of the paper, it would appear that elevating the temperatures at various pressures is productive of nearly uniform thermal gain. As pressures are raised at a constant temperature, Rankine efficiency increases rapidly at the lower station pressures, but the rate of increase becomes less at higher pressures.

Suppose it were decided to raise temperatures from 700 to 1000 deg. fahr., the question is, would there be an added inducement for using higher pressures? It appears that theoretically the gain due to increasing the temperature is fairly uniform for the various pressures, but with no special encouragement for extra high pressures.

The Rankine-cycle gains for various pressures up to 1200 lb., together with exhaust-steam conditions for a prime mover of 80

[1] Editorial staff, *Power*, New York, N. Y. Mem. A.S.M.E.

per cent Rankine-cycle-ratio efficiency, are given in Table 1. Accuracy here is intended to be sufficient for illustrative purposes only.

The temperature increase from 700 to 1000 deg. fahr. gives better relative results (increase in Rankine efficiency) at low pressures. The main gain theoretically from higher temperature and pressure appears to be in the Rankine efficiency due to increased pressure rather than relative gain due to a 300-deg. fahr. temperature rise. Practically, the moisture content in the exhaust is about halved at higher pressures by the temperature increase, indicating better relative gains by reduced friction at this end. Stage bleeding tends to be somewhat more beneficial at higher pressures.

Taking the gain due to friction as about equal to that of cycle efficiency where moisture is encountered at 300 lb. per sq. in., and somewhat less at lower pressures, with less moisture, a rough idea of the total benefit of temperature increases may be obtained from the last column of Table 1. In choosing between 700 and 1000 deg. fahr. temperature at the higher pressure, the theoretical gains do not appear impressive, but actual gains may be more favorable.

Temperature elevation brings a host of complications in its wake. Although the tensile or similar strength of materials may be singled out as a criterion for service, the effect on yield point is of equal or greater importance, since this drops more rapidly in some of the more familiar materials. Elasticity tends to give way to a plastic condition. Also the ability to take up or compensate for strains due to expansion requires elasticity primarily, so that this property for some services is highly desirable. Operating troubles owing to crystal growth, such as that exemplified in binding or distortion of valves, expansional distortions of the rotating or stationary prime-mover elements, changes in critical vibrating speeds of small elements, deterioration of metallic structure, etc., have been brought to attention in the past. As a result, preparations for temperature advances can now be made with a much better assurance of success than was possible a short time ago. Methods of testing materials, and manufacturing operations and design in general, have undergone appreciable changes in connection with advanced steam conditions.

The superimposed high-pressure installation, taken by itself, seems somewhat better suited for advances in pressure and temperature than the usual station taken as a whole. The high-pressure equipment of the type described in the paper is considered merely as being added to an existing plant. Here the regular load may be carried by the lower-pressure units, and shutdowns to the high-pressure plant ordinarily should not interrupt station service to the same extent as in the usual type of low-

pressure station. Besides, the high-pressure plant is not encumbered by condensing equipment, and in all probability may carry whatever load best meets its requirements, leaving the surges to be handled in large measure by the lower-pressure units.

TABLE 1 GAIN RESULTING FROM RAISING THE INITIAL TEMPERATURE FROM 700 TO 1000 DEG. FAHR. AS PRESSURES ARE INCREASED

Pressure, lb. per sq. in.	Temp. total, deg. fahr.	Rankine cycle eff., 80-deg. water	Increase Rank. eff. in per cent of eff. at 700 deg. fahr.	Per cent work in region of moisture, turb. eff. 80 per cent	Quality of steam at exhaust, per cent dryness	Estimated total gain in eff. by 300-deg fahr. increase, per cent
100	700	30.7		16	96	
100	1000	33.7	9.8	None	40 deg. superheat	11
200	700	33.7		27	92	
200	1000	36.2	7.4	None	15 deg. superheat	8
300	700	35.4	7.3	32	90	
300	1000	38.1		4.9	98	8
400	700	26.7		38	89	
400	1000	39.2	7.0	8.6	97	9
600	700	38.4		45	86	
600	1000	40.6	5.7	13	95	11
800	700	39.5		56	82	
800	1000	41.8	5.83	17	93	12
1200	700	40.7		60	80	
1200	1000	43.2	6.1	24	91	13

J. F. JOHNSON.[1] The thermal gain resulting from the use of high-pressure boilers and turbines as at the Weymouth station is readily determinable with reasonable accuracy by calculation. The definition of its influence on the station economics is more important and more involved. Its general adoption for improving the efficiency of existing stations is not probable.

The present rates of growth of generating capacity required by public-service companies, and of improvement in economic design of generating equipment, are both so rapid that existing stations operating at relatively high load factors become very quickly relegated to relatively low load factors and quick-demand service because newer stations required for increased capacity are economically superior in many respects. Higher investment cost due to replacement of existing boiler-room equipment and the increase in operating complications are also important factors unfavorable to its use in the majority of cases.

Other and more promising means for improvement are available. These include modifications to improve the efficiency and durability of furnaces, increase in operating steam pressure and temperature allowable with existing major equipment, and maximum practicable use of the regenerative cycle in connection with the heating of boiler feedwater and furnace air. Marked improvement in the efficiency of existing stations in which regenerative

[1] General Engineer, Westinghouse Elec. & Mfg Co., South Philadelphia Sub-Station, Philadelphia, Pa. Mem. A.S.M.E.

heating has not been provided may be made by the installation of an additional turbine unit designed for highest efficiency when extracting from it at the desired number of stages steam in sufficient quantity to heat feedwater for the entire station, and possibly also to operate evaporators and to preheat air for the furnaces.

One power company in New England has very materially reduced the coal rate of one of its stations by this method. Complete calculations and estimates for a similar installation in one of the large eastern stations have been made and it is probable that the method will be adopted. Here the quantities of steam extracted for feedwater heating were so large that when this turbine was generating 33,380 kw. only 188,900 lb. per hour, equal to 5.7 lb. per kw-hr., were rejected to the condenser.

Operating experience with large-capacity turbines with steam conditions as high as 350 lb. pressure and 700 deg. fahr. temperature at the throttle has been sufficient to warrant the entire removal of doubt as to its economic success. And the present satisfactory operation of stations with 550 lb. pressure and 725 deg. fahr. temperature indicates a similar status for these conditions.

For industrial plants requiring large quantities of steam for process work, the generation of this steam at higher pressure and its expansion through turbines to the process pressure offers possibilities of by-product power at attractive cost if the steam quantities are sufficiently large and the load factor sufficiently high. About eight years ago calculations and estimates were made for a 375-lb. installation involving 3000- to 5000-kw. turbines, exhausting at 150-lb. per sq. in. pressure, but the project was cancelled because of lack of confidence in the reliability of high-pressure equipment by the management.

In such installations the generation of steam at pressure as high as 600 lb. per sq. in. is more attractive from the efficiency standpoint than in pure power-generating stations, because the detriment caused by moisture in the low-pressure stages is not present. Full benefit of the increased heat drop is therefore obtained without the expense of reheating.

As to the mechanical design of turbines for high pressure and high temperature, there is at present no indication that turbines cannot be built to operate satisfactorily with steam at the highest pressure and temperature for which satisfactory boilers and piping can be built. Physical characteristics of metals available are the limiting feature for both, and the boiler and valve designers have the more difficult task.

ALFRED IDDLES.[1] The writer believes that many are interested in the particular part of the paper that mentioned the use of a

[1] Chief Power Engineer, Day & Zimmermann, Inc., Philadelphia, Pa. Mem. A.S.M.E.

high-pressure turbine operating with high back pressure, as might particularly be applied to industrial plants. Plants of an industrial nature which were built fifteen to twenty years ago normally had a boiler pressure of 150 lb. per sq. in., operating with saturated steam. Where process steam was usable, it was common practice to use a reciprocating engine exhausting at from 3- to 5-lb. per sq. in. pressure and to use that steam during the winter for heating, and during the summer for such process work as could use low-pressure steam. Today it is not at all impossible to use steam at 250-lb. per sq. in. pressure, with some superheat, and many of these old boiler plants, built twenty years ago, now must be revised. They can very easily be revised by the installation of boilers to operate at the higher pressure. In the old plants the drop in pressure is from 150 to 3 lb. per sq. in., but with the new installation 250-lb. per sq. in. pressure, with 100 deg. fahr. superheat, is satisfactory. If the drop is only to 25-lb. per sq. in. gage pressure there still is a higher total pressure drop with which to generate power. Therefore, there is 25-lb. steam available for distribution. It can be used for process purposes for which 3-lb. steam cannot be used. Transmitted in small pipes, with the present turbine design, the 25-lb. steam will have superheat, which again is to the advantage of this proposition for the reason that there will be fewer condensation losses in the piping.

One or two installations are now being very seriously considered in which the boilers will operate at 600-lb. per sq. in. pressure with 150 to 250 deg. fahr. of superheat used through the turbine. The latter will exhaust at 150 lb. per sq. in. That condition can be brought about by the requirement of this particular industry for 150-lb. steam instead of 25-lb. Those are possibilities that are easily brought about by reason of our progress in the design of the boiler plant. The 250-lb. plant will cost but little more than a 150-lb. one, and the 600-lb. plant would cost little more than the 250-lb. plant.

A feature that must be considered in order to take the greatest advantage of the pressure and superheat available is the uses to which the steam will be put. In a textile plant, wherein bleaching, washing, dyeing, and drying had to be done, two non-condensing turbines and one condensing turbine were installed, operating in parallel. In that connection it is well to study the load curves, for the reason that the use of process steam and the use of electric power almost never coincide. Therefore, there must be some provision for economically producing electric current when process steam is not needed. In the present case it is taken care of by the two non-condensing turbines operating at a back pressure of 15 lb. per sq. in. with a primary pressure of 175 lb., operating in parallel with a condensing turbine.

Another installation was a textile plant in which 50-lb. steam was necessary for process purposes. In both these plants central-

station energy was not available. The electrical energy required when process steam was not needed was furnished by a condensing turbine, with a non-condensing turbine operating in parallel. The load which each would take was so determined that exhaust steam at 50-lb. per sq. in. pressure was produced and no more.

E. W. Norris.[1] In designing the Weymouth station at Boston it was kept in mind that ultra-high-pressure equipment is perhaps a bit unproved, and one is not quite sure whether it will be used exclusively for base loads or be adaptable to peak-load operation. For that reason, in order to take advantage of base-load operation for initial work, and yet provide for peak-load operation later on, the high-pressure plant was entirely separated from the normal-pressure plant.

The power equipment consists essentially of two 350-lb. per sq. in. main turbines, of 30,000 kw. capacity, with complete boiler equipment and a 1200-lb. per sq. in. turbine and boiler designed to exhaust into the 350-lb. main steam header. In this case the 1200-lb. plant can be used to generate about 15,000 kw., 3000 in the high-pressure unit and 12,000 in the normal-pressure or 350-lb. unit.

This system was developed primarily to obtain great flexibility. Either the 350-lb. per sq. in. plant, or the 1200-lb. plant can be developed as experience indicates. In either case the boilers will occupy the same space in the building, and, the high-pressure turbine being very small, additional turbines can be taken care of easily in the auxiliary-machinery bay. In this way, if found in years to come that the high-pressure plant is suitable for peak-load operation as well as for base-load operation, there will be no interference between the 1200-lb. and the 350-lb. plants. The higher-pressure unit can be expanded to take care of any portion of the load thought desirable.

With respect to the development in boilers and turbine, it should be pointed out that boiler practice is rapidly getting to a point where but little further advance can be expected. We are working, in new plants, presumably above 80 per cent efficiency; in some cases above 85 per cent. These efficiencies do not leave a great deal of margin for improvement. On the other hand, the efficiency of the turbine cycle is extremely low, and developments there have a very wide field of possibility. In that respect the regenerative cycle is a notable step in the right direction.

The Author. Professor Christie remarks that reconstructing the boiler room in old plants along modern lines would involve too great an expense in order to make it practical, and therefore

[1] Engineer, Mechanical Division, Stone & Webster, Inc., Boston, Mass. Mem. A.S.M.E.

proposes to use these plants for peak-load or standby service. It is obvious that many of the older plants are not suitable for the reconstruction as proposed, but at the same time there are some comparatively new efficient plants operating at steam pressures of about 250 lb. where such reconstruction may be highly economical and at the same time practical. To hold old low-pressure power plants for standby or peak-load service as suggested can be done, of course, but the indications seem to be that those plants will be completely shut down and kept in condition for emergency operation only. Presumably, a strong reason for keeping the older plants intact is the fact that in most states rate-making bodies exist, and therefore the companies are reluctant to destroy the capital invested on which a return has to be earned.

Mr. Thompson's comments with Table No. 1 showing the gain in efficiency from 700 deg. fahr. to 1000 deg. fahr. at various pressures are very interesting and form a valuable contribution to the discussion.

Mr. Johnson's comments with reference to one method of increasing the efficiency of existing plants are interesting, but its use does not change the conditions under which the installing of high-back-pressure turbines with boilers is proposed, as such turbines can be satisfactorily installed with this system also and show a material increase in efficiency.

Mr. Iddles' comments on high-pressure turbines for industrial plants are in line with work that has been done. With the higher pressures and temperatures made available, the high-back-pressure turbines for industrial plants will probably become more standardized. For certain sizes and conditions, fairly well-standardized turbines can now be purchased from the manufacturer.

In Mr. Morris' comments the system adopted at the Weymouth station is described. This conforms with the one proposed by the author, namely, a high-pressure turbine with or without reheating the exhaust steam before it passes into the main steam turbine. In all probability, if the steam pressures and temperatures are further increased, this system of separate high-back-pressure turbines will be still further developed, as the construction indicates a practical way for utilizing the high pressures and temperatures economically.

No. 1962

ANALYSIS OF POWER-PLANT PERFORMANCE BASED ON THE SECOND LAW OF THERMODYNAMICS

By Wm. L. DeBaufre,[1] Lincoln, Neb.
Member of the Society

The author shows that the commonly used heat balance based on the first law of thermodynamics does not allocate the losses in a power plant in such a manner as to reveal the best points of attack on the problem of improving power-plant economy. The application of the second law gives a quite different statement of the losses of energy from the coal pile to the stack and engine exhaust. As an example, the performance of a locomotive is analyzed by both the first- and second-law methods.

IT IS customary to analyze power-plant performance by preparing a heat balance based on the first law of thermodynamics to show the distribution of the heat of combustion of the fuel. The object of operating a power plant, however, is not to distribute heat but to convert heat into mechanical work. An analysis based upon the second law of thermodynamics is therefore much more enlightening than the usual first-law heat balance, because such an analysis shows the lost opportunity for converting heat into mechanical work due to the imperfections in each stage of the process.

2 Before presenting such an analysis, some remarks will be made upon the second law of thermodynamics. These remarks are believed to be in order since engineers apparently have such a hazy idea of the meaning of the second law. The first law can be stated in simple words, such as:

> Heat and mechanical work are different forms of energy, and when one is converted into the other, the ratio is 778 ft-lb. to 1 B.t.u.

The same brevity has been attempted in stating the second law, with the result that this law has generally been a jingle of words rather than a working tool. In order to present a definite con-

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ception of the second law which will enable it to be used as the basis for an analysis of power-plant performance, some very elementary principles will first be stated.

3 To do mechanical work it is necessary to exert a force through a distance, because work is so defined. The addition of heat to a solid body or to a fluid generally expands the substance, even against considerable resistance. The mechanical work thus performed is equivalent to a greater or less proportion of the heat added. With isothermal expansion of air under moderate pressures, for example, practically 100 per cent of the heat added is converted into mechanical work. The idea that such a thing is impossible is one widespread misconception of the second law. The second law of thermodynamics, however, does not apply to this case, but only to cases where the same medium must be used over and over again.

4 Under this limitation it is impossible on this earth to convert into mechanical work 100 per cent of the heat supplied for the reason that some heat must be rejected by the working substance to contract it in order that it may again be expanded in using it the second time. To expand the working substance there must be a "source of heat" at a higher temperature than the working substance; and to contract the working substance there must be a "source of cold" at a lower temperature. By bringing the working substance alternately into contact with the "source of heat" and the "source of cold," mechanical work may be performed equivalent to the difference between the heat supplied from the source of heat and that rejected to the source of cold.

5 As shown by Carnot, there is a certain maximum ratio of the heat converted into mechanical work to that supplied from the source of heat, which ratio is the efficiency of a perfect engine. This ratio is a function of the upper and lower temperatures only, and is independent of the nature of the working substance used in the perfect engine. The Carnot cycle shows theoretically how this perfect engine must operate. The Stirling cycle shows another method of attaining the maximum efficiency, but a working substance with certain specific-heat relations is required.

6 From textbooks on thermodynamics, the following mathematical expressions may be obtained for the maximum possible conversion of heat into mechanical work and the corresponding minimum rejection of heat to the source of cold with the Carnot cycle. Calling these quantities the "available energy" and "unavailable energy," respectively, we have

$$\text{Available energy, } dW = \frac{T-T_0}{T}\, dQ = (T-T_0)dS. \quad . \quad [1]$$

$$\text{Unavailable energy, } dQ_0 = \frac{T_0}{T}\, dQ = T_0 dS \quad . \quad . \quad . \quad . \quad . \quad . \quad [2]$$

where T = absolute temperature of the source of heat
T_0 = absolute temperature of the source of cold
dQ = heat supplied at temperature T to the working substance in a perfect engine
dQ_0 = heat rejected at temperature T_0 by the working substance in a perfect engine
dW = heat converted into mechanical work by a perfect engine, and
$dS = dQ/T$, by definition the change in entropy of the material source of heat.

7 These mathematical relations express the second law of thermodynamics for which the following statement is proposed:

> When a medium is used to convert heat into mechanical work by operating in a closed cycle between an upper absolute temperature T and a lower absolute temperature T_0, only a portion of the heat supplied at the upper temperature can be converted into mechanical work when the lower temperature is above absolute zero; and the relation between the heat supplied at the upper temperature dQ, the maximum quantity available for conversion into mechanical work dW, and the minimum quantity rejected at the lower temperature dQ_0, is given by

$$\frac{dQ}{T} = \frac{dW}{T-T_0} = \frac{dQ_0}{T_0} \quad \ldots\ldots\ldots [3]$$

This relation also applies to the reverse operation, but we are not at present concerned with refrigerating machines.

8 Based on this statement of the second law of thermodynamics, it is possible to analyze the performance of a power plant step by step and tabulate the losses of available energy occurring in each stage of the process. In explaining this analysis to classes in thermodynamics, the greatest difficulty encountered has been to get the students to understand that the perfect engine of Carnot or Stirling is applied simply as a measuring rod at each step of the process, none of the fluids in the power plant analyzed being the medium used in the perfect engine. The underlying conceptions for this analysis are: (*a*) A perfect engine is assumed to be operated by heat supplied from each of the actual fluids in turn, furnace gases, steam, etc., at each step in the process; (*b*) The temperature of heat supply will vary according to the change in temperature which would be experienced by the fluid in question giving up its heat to the perfect engine; (*c*) The temperature of exhaust of the perfect engine will be constant and equal to that of the atmosphere or of the water available for condensing purposes.

9 In applying relations [**1**] and [**2**] to determine the available energy and the unavailable energy at any step in the process, the calculations are very simple when steam is the source of heat to

the perfect engine because the entropies of steam are tabulated. In this case, by integration of [2] we have

$$\text{Unavailable energy, } Q_0 = T_0(S_1 - S_2) \quad [4]$$

$$\text{Available energy, } W = Q - Q_0 \quad [5]$$

where S_1 and S_2 are the entropies of the steam at the beginning and end, respectively, of the withdrawal of heat which is supplied to the perfect engine.

10 For the furnace gases, of which the entropies are not known, Formula [2] may be integrated by using the relation

$$dQ = (a + bT)dT \quad [6]$$

where a and b are the specific-heat coefficients. Relation [6] may be integrated to obtain the heat supplied by the furnace gases in cooling from T_1 to T_2, namely,

$$\text{Energy supplied, } Q = a(T_1 - T_2) + \frac{b}{2}(T_1^2 - T_2^2) \quad [7]$$

$$= \left[a + \frac{b}{2}(T_1 + T_2)\right](T_1 - T_2) \quad [7a]$$

Integrating [2] with the aid of [6], we have

$$\text{Unavailable energy, } Q_0 = aT_0 \log_e \frac{T_1}{T_2} + bT_0(T_1 - T_2) \quad [8]$$

$$\text{Available energy, } W = Q - Q_0 \quad [9]$$

When the specific heat may be assumed constant with a sufficient degree of accuracy, Formulas [7] and [9] reduce to

$$\text{Energy supplied, } Q = a(T_1 - T_2) \quad [10]$$

$$\text{Unavailable energy, } Q_0 = aT_0 \log_e \frac{T_1}{T_2} \quad [11]$$

$$\text{Available energy, } W = Q - Q_0 \quad [12]$$

11 The calculations for unavailable energy are simpler than for available energy; and as the change in unavailable energy for any stage of the process must equal the change in available energy, the former is generally calculated in preference to the latter.

12 As an example of the application of the second law of thermodynamics enunciated above, the performance of a locomotive will be analyzed, the following data being taken from P. R. R. Bulletin 28, for test No. 3925 of Mikado type locomotive, Pennsylvania Class L1s:

Heating value of coal	14,044 B.t.u. per lb. of dry coal
Temperature in firebox	2490 deg. fahr.
Temperature in smokebox	494 deg. fahr.
Temperature of feedwater	70 deg. fahr.
Temperature of superheated steam in branch pipe	555 deg. fahr.
Temperature in exhaust pipe	263.1 deg. fahr.
Atmospheric pressure	14.2 lb. per sq. in.
Steam pressure in branch pipe	193.7 lb. gage, say, 208 lb. per sq. in. abs.

Pressure in exhaust passage.....8.2 lb. gage, or 22.4 lb. per sq. in. abs.
Dry coal fired per sq. ft. of grate surface.....86.6 lb. per hr.
Steam to engines per lb. of dry coal fired.....7.03 lb.
Steam per indicated horsepower.....18.2 lb. per hr.

13 The calculations for the ordinary heat balance based on the first law will first be made:

1297.4 B.t.u. per lb. total heat of superheated steam, from steam tables

38.1 B.t.u. per lb. heat in feedwater, from steam tables

$1297.4 - 38.1 = 1259.3$ B.t.u. to produce one lb. of steam

$1259.3 \times 7.03 = 8853$ B.t.u. transmitted to the steam per lb. of dry coal.

14 For a temperature of 2490 deg. fahr. in the fuel bed and a heating-surface temperature of, say, 390 deg. fahr., we have

$$1600\left[\left(\frac{460+2490}{1000}\right)^4 - \left(\frac{460+390}{1000}\right)^4\right] =$$

$1600(75.74 - 0.52) = 120{,}350$ B.t.u. per hr. per sq. ft. of grate surface radiated from the fuel bed to the heating surface

$$\frac{120{,}350}{86.6} = 1390$$ B.t.u. per lb. of dry coal reaches the heating surface by radiation

$8853 - 1390 = 7463$ B.t.u. per lb. of dry coal reaches the heating surface by convection.

15 Assume the specific heat of the products of combustion to be given by $0.24 + 0.00002\,T$, where T is the absolute temperature in degrees fahrenheit. The mean specific heat between T_1 and T_2 is then given by $0.24 + 0.00001\,(T_1 + T_2)$ B.t.u. per lb., or according to Formula [**7a**],

$[0.24 + 0.00001(460 + 2490 + 460 + 494)](2490 - 494) =$
$0.279 \times 1996 = 556.9$ B.t.u. per lb. of products of combustion given up to heating surface

$$\frac{7463}{556.9} = 13.40$$ lb. of products of combustion per lb. of dry coal

$[0.24 + 0.00001(460 + 494 + 460 + 70)](494 - 70) =$
$0.225 \times 424 = 108.1$ B.t.u. in stack gases per lb. of gases

$108.1 \times 13.40 = 1449$ B.t.u. in stack gases per lb. of dry coal

$8853 + 1449 = 10{,}302$ B.t.u. per lb. of dry coal produced by combustion

$14{,}044 - 10{,}302 = 3742$ B.t.u. per lb. of dry coal lost by unburnt fuel, incomplete combustion, etc.

$$\frac{18.2}{7.03} = 2.59$$ lb. of dry coal per hr. burned per indicated horsepower

$$\frac{2546}{2.59} = 983$$ B.t.u. converted into mechanical work per lb. of dry coal

$8853-983 = 7870$ B.t.u. in exhaust steam per lb. of dry coal

$\frac{7870}{7.03} = 1119$ B.t.u. per lb. of exhaust steam above 70 deg. fahr.

$1119+38 = 1157$ B.t.u. per lb. total heat of exhaust steam

$202.2+x\times 957.8 = 1157$, whence $x = 0.997$-dryness fraction of exhaust steam.

NOTE: The temperature in the exhaust pipe was 263.1 deg. fahr. by measurement, thus indicating 29 deg. of superheat. It is therefore probable that the superheat in the steam initially was greater than measured. To be consistent, however, the exhaust steam will be assumed to have the quality calculated.

FIRST-LAW BALANCE

	B.t.u.	Per cent
Heat lost by unburnt coal, etc.....	3,742	26.7
Heat lost in stack gases.....	1,449	10.3
Heat converted into work.....	983	7.0
Heat rejected in exhaust steam	7,870	56.0
Heating value of dry coal.....	14,044	100.0

16 Calculations for the second-law balance will now be made of available energy and unavailable energy at each step of the process in order to determine the loss in available energy, or gain in unavailable energy, during each stage. We shall assume the lower temperature to be that of the feedwater, 70 deg. fahr. (530 deg. abs.).

17 During the first stage of the process, namely, the combustion of the fuel, the 3742 B.t.u. per lb. of dry coal not produced by reason of unburnt coal, incomplete combustion, etc., is evidently unavailable for conversion into mechanical work.

18 Of the 10,302 B.t.u. per lb. of dry coal resulting from combustion, 1390 B.t.u. are radiated from the fuel bed at the constant temperature of 2490 deg. fahr. From Formula [**2**],

$1390\times\frac{460+70}{460+2490} = 1390\times 0.1797 = 250$ B.t.u. per lb. of dry coal (unavailable portion of radiant heat).

$1390-250 = 1140$ B.t.u. per lb. of dry coal (available portion of radiant heat).

19 The remainder of the 10,302 B.t.u., namely, 8912 B.t.u. per lb. of dry coal, reaches the heating surfaces by convection of the furnace gases. That is, this heat leaves the furnace in 13.40 lb. of products of combustion at 2490 deg. fahr. (2950 deg. abs.). Substituting in Formula [8] under the assumption that these gases are cooled to 70 deg. fahr. by imparting their heat to operate a perfect engine, we have

$$530\left[0.24\log_e\frac{2950}{530}+0.00002(2950-530)\right] =$$

$530\times 0.4604 = 244.0$ B.t.u. unavailable energy per lb. of furnace gases

$13.40 \times 244.0 = 3270$ B.t.u. unavailable energy in furnace gases per lb. of dry coal

$8912 - 3270 = 5642$ B.t.u. per lb. of dry coal available in firebox gases for transformation into mechanical work by a perfect heat engine.

20 These products of combustion escape to the smokebox at a temperature of 494 deg. fahr. (954 deg. abs.) and have a sensible heat above 70 deg. fahr. of 1449 B.t.u. per lb. of dry coal. Substituting in Formula [8],

$$530 \left[0.24 \log_e \frac{954}{530} + 0.00002(494 - 70) \right] = 530 \times 0.1496$$

$= 79.29$ B.t.u. unavailable per lb. of smokebox gases

$13.40 \times 79.29 = 1062$ B.t.u. per lb. of dry coal unavailable energy in smokebox gases

$1449 - 1062 = 387$ B.t.u. per lb. of dry coal available energy in smokebox gases, but lost with these gases passing out the stack.

21 Of the 10,302 B.t.u. per lb. of dry coal resulting from combustion, therefore,

$1140 + 5642 = 6782$ B.t.u. per lb. of dry coal are available for conversion into mechanical work by a perfect engine.

But 387 B.t.u. of this available energy are lost with the escape of the products of combustion out of the stack. Hence, of the heat transferred to the steam,

$6782 - 387 = 6395$ B.t.u. per lb. of dry coal were available for conversion into mechanical work by a perfect engine before the transmission of heat through the heating surface.

22 To find the availability of the 8853 B.t.u. per lb. of dry coal transmitted through the heating surface after its transmission, we obtain from steam tables:

1.6486 entropy per lb. of superheated steam and

0.0746 entropy per lb. of water at 70 deg. fahr.

Substituting in Formula [4],

$530(1.6486 - 0.0746) = 834.2$ B.t.u. unavailable per lb. of steam

$7.03 \times 834.2 = 5864$ B.t.u. per lb. of dry coal unavailable energy in the superheated steam

$8853 - 5864 = 2989$ B.t.u. per lb. of dry coal are available for conversion into mechanical work by a perfect engine after transmission through the heating surface

$6395 - 2989 = 3406$ B.t.u. per lb. of dry coal loss of available energy due to heat transfer in the boiler.

23 Now, for the exhaust steam, we have from the steam tables corresponding to the quality previously determined,

$1.7293 - 0.003 \times 1.3809 = 1.7252$ entropy per lb.

Substituting in Formula [4],

$530(1.7252 - 0.0746) = 874.8$ B.t.u. unavailable per lb. of steam

$7.03 \times 874.8 = 6150$ B.t.u. per lb. of dry coal unavailable energy in exhaust steam.

Since the exhaust steam contains 7870 B.t.u. per lb. of dry coal (above water at 70 deg. fahr.) according to previous calculations, we have

$7870 - 6150 = 1720$ B.t.u. per lb. of dry coal available for conversion into mechanical work by a perfect engine, but carried out by the exhaust steam.

24 Since 2989 B.t.u. per lb. of dry coal were available in the steam entering the engine and 983 B.t.u. per lb. of dry coal are equivalent to the work actually done, we have

$2989 - 983 - 1720 = 286$ B.t.u. per lb. of dry coal loss of available energy due to imperfections in the engine cylinders.

25 It may be noted that if the engine cylinders had been less imperfect in their operation, the available energy lost in the exhaust steam would have been somewhat less. The possibility of improvement in the engine cylinders is therefore somewhat greater than the figure 286 would indicate.

26 Summing up the losses of available energy, we obtain

SECOND-LAW BALANCE

	B. t. u.	Per cent
Unavailable due to unburnt fuel, etc.	3,742	26.6
Unavailable part of radiant energy	250	1.8
Unavailable energy in firebox gases	3,270	23.3
Available energy lost in smokebox gases	387	2.8
Available energy lost by heat transfer	3,406	24.2
Heat transformed into work	983	7.0
Available energy lost by imperfections in cylinders	286	2.0
Available energy lost in exhaust	1,720	12.3
Heating value of dry coal	14,044	100.0

27 Comparing the first-law balance with the second-law balance, we find some items appearing in the latter which do not occur in the former, while other items are greatly altered in their relative proportions. Thus, one of the largest losses appearing in the second-law balance is that due to heat transfer in the boiler, amounting to nearly one-quarter of the heating value of the fuel. This is the loss that is now being successfully attacked and reduced by the use of higher steam pressures and other fluids than water, such as mercury.

28 The loss in the exhaust steam appears reduced from over one-half of the heating value of the fuel in the first-law balance to less than one-eighth in the second-law balance, which is its correct relative importance. In an analysis of a condensing plant the loss in the exhaust is a still lower portion of the heating value of the fuel, thus indicating the gain due to the use of condensers.

29 In calculating the second-law losses during combustion, various assumptions might be made in order to prorate these losses among unburnt fuel, incomplete combustion, excess air, etc., and in certain cases it will be desirable to attempt such a division. The order in which the calculations are made, however, will affect the prorating. In many cases the simple expedient adopted in these calculations will suffice.

30 In presenting the second-law analysis for the first time, a simple case has been chosen for illustration, namely, a locomotive. No great difficulties, however, will be encountered in applying this analysis to more complicated power systems. While somewhat more complicated calculations are required for the second-law balance than for the usual first-law balance, it is believed that the facts revealed thereby will more than justify its extended use.

31 In closing, the author wishes to acknowledge his indebtedness for the general idea of preparing a second-law balance to Dr. R. C. Tolman, of the California Institute of Technology.

DISCUSSION

T. E. BUTTERFIELD.[1] While it is true that, for some years, engineers have been more seriously considering the availability of heat energy, or conservation of available heat energy, yet this increased interest has not been adequately reflected in testing codes or in power-plant test calculations. A century ago Carnot showed that only a fraction of the heat energy released by the combustion of a fuel could be turned into work, even by an ideally perfect engine. This fraction, or the available energy, is further diminished or partly dissipated wherever there is an unbalanced temperature drop or unbalanced pressure drop.

The author places a needed emphasis on a class of calculations which, though taught in courses in thermodynamics, and fundamental in any careful study of the transformation of heat into work, are probably not sufficiently stressed, even at the present day, in power-plant work.

The writer believes that the discrepancy in the reported water rate of 18.2 lb. per hp-hr., with the entering steam at 208 lb. and 555 deg. fahr. and exhaust steam at 22 lb. and 263 deg., is more probably due to an error in measuring the indicated horsepower than to one in the measurement of the exhaust temperature. With

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no heat losses from the cylinder walls, the work done per pound of steam, expressed in heat units, is equal to the difference of the total heats of the steam entering and leaving the cylinder, or to 1293.8 − 1172.3 = 121.5 B.t.u. This corresponds to a water rate of 2545 ÷ 121.5, or 20.8 lb. of steam per hp-hr., instead of 18.2 as reported. A proportionate error in the indicated horsepower appears probable. To make possible a closer comparison with the author's figures the writer in the following calculations has assumed that the discrepancy was due to an incorrect determination of the temperature of the exhaust steam.

An error is involved in the assumption that all heat given up by the furnace gases in cooling to the stack temperature of 494 deg. fahr. is transmitted to the steam. Part of this heat enters the steam and leaves again in the form of radiation and conduction from the exterior surface of the boiler. The same retransmission of heat occurs in the case of part of the radiation from the incandescent surface of the fuel bed to the interior surface of the firebox. The heat given up by the stack gases in cooling is therefore greater than the heat absorbed by the steam. Therefore, more pounds of furnace gases are circulated than calculated by the author. In the calculations following, however, the assumption of zero radiation loss is followed, in order to permit a closer comparison with figures of the paper.

The formula for the specific heat of furnace gases given in Par. 10 is in error in the use of absolute temperatures instead of temperatures above fahrenheit zero. This has led to slight errors in the calculated values of heat absorption and of entropy change per pound of gas.

Basic Figures. The following preliminary calculations are given because with their aid it will be much easier to follow the main calculations, and also because the Marks and Davis tables, which the writer has used, differ slightly from the tables used in the calculations of the paper.

Steam at 208 lb. per sq. in. and 555 deg. fahr.

Total heat = 1293.8 B.t.u. per lb.

Entropy = 1.6455 per lb.

Heat supplied = 1293.8 − 38.1 = 1255.7 B.t.u. per lb.

Unavailable energy, above 70 deg. = (1.6455 − 0.0745)530 = 833 B.t.u. per lb.

Available energy = 1255.7 − 833 = 422.7 B.t.u. per lb.

Steam at 22 lb. and 0.996 quality.

Total heat = 1158.0 − (0.004 × 956.7) = 1154.0 B.t.u. per lb.

Entropy = 1.7241 − (0.004 × 1.3811) = 1.7186 per lb.

Unavailable energy = (1.7186 − 0.0745) × 530 = 871.4 B.t.u. per lb. above water at 70 deg. fahr.

Available energy = 1154.0 − 38.1 − 871.4 = 244.5 B.t.u. per lb.

Useful work for steam expanding from 208 lb. and 555 deg. fahr. = 1293.8 − 1154.0 = 139.8 B.t.u. per lb.

2545 ÷ 139.8 = 18.2 lb. per hp-hr.

Furnace gases at 2490 *deg. fahr.*

Mean specific heat from 70 deg. to 2490 deg. fahr. = 0.24 + 0.00001 × (2490 + 70) = 0.2656

Heat loss from 2490 deg. to 70 deg. = 0.2656(2490 − 70) = 642.7 B.t.u. per lb.

Entropy change from 2490 to 70 deg. fahr. = $0.2656 \log_e \frac{2950}{530}$ = 0.4549 per lb.

Unavailable energy at 2490 deg. fahr. = 0.4549 × 530 = 241.1 B.t.u. per lb.

Available energy = 642.7 − 241.1 = 401.6 B.t.u. per lb.

Cooling from 2490 *deg. to* 494 *deg. fahr.*

Mean specific heat = 0.24 + 0.00001(2490 + 494) = 0.2698

Heat loss = 0.2698(2490 − 494) = 538.6 B.t.u. per lb.

Entropy loss = $0.2698 \log_e \frac{2950}{954}$ = 0.3047 per lb.

Unavailable energy = 0.3047 × 530 = 161.5 B.t.u. per lb. of gas

Available energy = 538.6 − 161.5 = 377.1 B.t.u. per lb. of gas

Cooling from 494 *deg. to* 70 *deg. fahr.*

Mean specific heat = 0.246

Entropy loss = $0.246 \log_e \frac{954}{530}$ = 0.1446 per lb.

Heat loss = 0.246(494 − 70) = 104.3 B.t.u. per lb.

Unavailable heat = 0.1446 × 530 = 76.6 B.t.u. per lb.

Available heat = 104.3 − 76.6 = 27.7 B.t.u. per lb.

Detailed Distribution of Losses of Available Heat Energy. The losses of available energy occur in the following ways:

1 In the transformation of chemical energy to radiant heat
2 During the radiation of heat from the glowing surface of the fuel to the water and steam in the boiler
3 In transformation of chemical energy to sensible heat of the furnace gases
4 During conduction of heat from the hot furnace gases to the cooler water and steam
5 During cooling of the stack gases, loss of available heat
6 Carried away by the stack gases, unavailable heat
7 In unburnt fuel
8 Available heat in the exhaust steam
9 Unavailable heat in the exhaust steam.

Finally there remains

10 Heat transferred to indicated work.

Heat Passing to Steam by Radiation. The total, 1390 B.t.u. per lb. of coal, will be taken from Par. 14 of the paper. Before trans-

mission, part of this heat is unavailable. The amount of initially unavailable heat is

$$(530 \div 2950)1390 = 250 \text{ B.t.u. per lb. of coal.}$$

It will be desirable to compare this 250 B.t.u. initially unavailable, with the amount of the same radiant energy that becomes unavailable on transmission to the steam. As all of the 1390 B.t.u. have been assumed to be used in making steam, the number of pounds of steam made will be

$$1390 \div 1255.7 = 1.107 \text{ lb. per lb. of coal.}$$

The unavailable heat in one pound of steam at 208 lb. and 555 deg. fahr. is 833 B.t.u. as found earlier in this discussion. Therefore, the unavailable heat in 1.107 lb. steam will be 922 B.t.u. per lb. of coal. The loss of available energy in passing the 1390 B.t.u. from the coal to the steam by radiation is $922 - 250 = 672$ B.t.u. per lb. of coal. This is due to the temperature difference between the coal, at 2490 deg. fahr., and the water and steam at 75 deg. to 555 deg. fahr.

Heat Passing to Steam by the Cooling of the Furnace Gases. Of the 7.03 lb. of steam generated per pound of coal, 1.107 lb. are supplied by the 1390 B.t.u. of radiation, leaving 5.923 lb. to be supplied by cooling the furnace gases from 2490 deg. to 494 deg. fahr. The heat absorbed is

$$5.923 \times 1255.7 = 7438 \text{ B.t.u. per lb. of coal.}$$

From the basic figures for furnace-gas cooling, given earlier, the number of pounds of furnace gases required to give out this amount of sensible heat is

$$7438 \div 538.6 = 13.81 \text{ lb. per lb. of coal.}$$

This ignores radiation loss, so that the actual amount will be much greater.

It is now evident that $161.5 \times 13.81 = 2230$ B.t.u. per lb. of coal out of the total 7438 would become unavailable when the gas absorbed the heat from the coal, leaving 5208 B.t.u. available. After transfer of this 7438 B.t.u. to the steam, the available energy received by the latter is $5.923 \times 422.7 = 2504$ B.t.u. per lb. of coal, giving an available energy loss equal to $5208 - 2504 = 2704$ B.t.u. per lb. of coal, due to the transfer of heat from the gas to the steam, with the large average temperature difference existing.

Stack-Gas Losses. As in the case of furnace-gas cooling, the cooling of stack gases is associated with the loss of available energy in the process of cooling, and also with the passing on or final disposition of a further amount of energy which became unavailable when the sensible heat in the stack gases was first imparted, that is, at the instant of heating. The available energy in the gases entering the stack is $13.81 \times 27.7 = 382$ B.t.u. per lb. of coal. The further amount of $13.81 \times 76.6 = 1059$ B.t.u. per lb. of coal is

unavailable heat in the gases entering the stack, but this amount of heat becomes initially unavailable on combustion.

Losses in Engine. As the steam reaches the cylinder at a pressure of 208 lb. per sq. in. and a temperature of 555 deg. fahr. it contains 422.7 B.t.u. per lb. of available energy. As it leaves the engine at a pressure of 22 lb. per sq. in. and with 0.996 quality, it contains 244.5 B.t.u. of available energy. The work done per pound of steam on the piston is 139.8 B.t.u. The available energy lost or wasted by the engine is $422.7-139.8-244.5 = 38.4$ B.t.u. per lb. of steam, or $7.03 \times 38.4 = 272$ B.t.u. per lb. of coal. The useful work done is $7.03 \times 139.8 = 983$ B.t.u. per lb. of coal.

Losses in Exhaust. As the exhaust steam leaves the cylinder, it contains, above water at 70 deg. fahr.:

Total heat: $1154.0-38.1 = 1115.9$ B.t.u. per lb. steam, or $1115.9 \times 7.03 = 7845$ B.t.u. per lb. of coal.

Available heat: 244.5 B.t.u. per lb. steam, or 1719 B.t.u. per lb. coal.

Summary. In Table 1, for each heat transfer, the corresponding loss is shown. The sum of 1, 3 and 5 shows that 25.22 per cent of

TABLE 1 SUMMARY OF LOSSES OF AVAILABLE ENERGY

	Entering			Leaving		Loss of available heat	
	Total heat passing	Not available	Available	Not available	Available	B. t. u. per lb coal	Per cent
1 Combustion [1]	....	....	....	....	....	250	1.78
2 Radiation	1390	250	1140	922	468	672	4.79
3 Combustion [1]	....	....	....	....	....	2230	15.90
4 Conduction ..	7438	2230	5208	4934	2504	2704	19.25
5 Combustion [1]	....	....	....	....	....	1059	7.54
6 Stack cooling..	1441	1059	382	1059	....	382	2.71
7 Unburnt fuel..	3776	....	3776	3776	....	3776	26.85
8 Engine	8828	5856	2972 [2]	6123	1719	270	1.90
10 Useful work ..	....	....	....	....	....	983	7.00
9 In exhaust	7847	6123	1719	7845	....	1719	12.28
						14,045	100.00

[1] Sum of 1, 3 and 5 = 25.22 per cent.
[2] Equal to 21.2 per cent.

the heat in the coal is wasted by the particular method of combustion used in the steam boiler. The boiler delivers to the engine in an available form only 21.2 per cent of the heat in the coal. Nearly twice the heat utilized by the engine is wasted in available heat delivered to the exhaust.

R. EKSERGIAN.[1] Accurate knowledge wherein thermal gains may be obtained in the refinement and general improvement now going on in steam-engineering development is based on a clear understanding of the limitations imposed. These limitations are, broadly,

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well-defined mechanical or constructive limitations for the particular apparatus, and the thermal limitations. The potentiality or availability of heat is the all-important basis for judging the real performance of heat engines, and the more we can differentiate the potential or available head *along the process*, the clearer will be the outline for possible improvement.

The author has given an interesting and valuable contribution of a simple application of the second law of thermodynamics in a heat balance along the process. Carnot's principle when applied between a constant-temperature source of heat and cold is easy to appreciate, since the effective head is simply measured by the difference between the extreme constant-temperature limits and is independent of the working substance, but when heat is added at varying temperatures each small quantity of heat so added has its own available head, depending upon the upper temperature of the heat added. That is, the second law should always be thought of as a differential process as stated in Equations [**1**] and [**2**]. The real significance of the entropy is then recognized as a coördinate with the temperature. Further, in the summation the total available energy is dependent not only upon the temperature limits, but also on how this energy is contained in the substance as shown by Equation [**6**]. Thus the real application of the second law is a different process.

Proceeding to the application of a locomotive test, the Stefan and Boltzmann formula for radiation is stated in terms of grate surface and as entirely independent of the receiving surface; i.e., the firebox heating surface. Though it is acknowledged that several authorities base the radiation on the grate, it appears inconsistent with test observations, since the amount of heat absorbed by the tubes would be too great. Of course, it is admitted that the effective surfaces subjected to radiation and the corresponding temperatures are rather indefinite, but it would seem, however, in a first approximation, if we regard the radiation based on the emissive surface rather than the receiving surface, that the real emissive surface for the long flames found in locomotive fireboxes would be an equivalent ideal tubular surface surrounding the flames. This would more closely approach the firebox surface rather than the grate surface. Thus the heat transmitted by radiation would be something in the order of three or four times that stated in the paper. The grate area of the locomotive considered was 70 sq. ft. Hence the total coal fired was 6060 lb. per hr. If 7463 B.t.u. were to be absorbed by convection in the tubes per pound of coal, it would mean a unit evaporation in the tubes (3715.7 sq. ft.) of $(6060 \times 7463)/3715.7 \times 1259.3) = 9.7$ lb. per sq. ft. of tube surface. The normal evaporation of these tubes was estimated at $(42{,}600/54{,}000) \times 9.3 = 7.3$ lb. at a firing rate of 86.6 lb. of coal per sq. ft. of grate per hr. The firebox evaporation for a heating surface 301.5 sq. ft. would be only $(6060 \times 1390)/(1259.3 \times 301.5) = 22.2$

lb. per sq. ft. of firebox surface, as against 40 to 55 lb. actual estimated in practice. Moreover, the temperature offers a crude check on the relative magnitude of the heat transferred by radiation. The equation for the firebox temperature, neglecting the small heat transferred by convection, may be stated as

$$nR\int_{70}^{t}(a+bT)dT\,\frac{A_e}{A_g}\,1600\left[\left(\frac{460+t}{1000}\right)^4-\left(\frac{460+t_s}{1000}\right)^4\right]=$$

$$14{,}000(1-CR)R$$

where n = pounds of products of combustion per lb. of coal
A_g = area of grate, sq. ft.
A_e = equivalent emissive surface, sq. ft.
t = flame temperature in firebox, deg. fahr.
t_s = temperature of steam, approximate, deg. fahr.
R = rate of firing, lb. per sq. ft. per hr.
C = constant depending upon ratio of grate to firebox surface, volume, etc.

Assuming 70 deg. fahr. for the initial temperature, 12 lb. of products of combustion per pound of coal actually fired, with the combustion efficiency of 80 per cent and a mean firebox temperature of 2490 deg. fahr., the ratio of A_e to A_g would be 2.31.

C. Harold Berry.[1] In studying the performance of a steam power plant by means of a first-law heat balance, one is struck by the very large loss in the exhaust, whether atmospheric or condensing. Those not familiar with the thermodynamic relations involved often suppose that this large loss is due to some failure of the engineer to take advantage of the possibilities. The author has given us an analysis which shows this loss in a manner more in keeping with the possibility of its reduction.

On the other hand, his analysis gives larger relative values for the losses in the boiler room, and this may prove misleading. It is true that the boiler develops no mechanical work. It is also true that, with the interposition of appropriate apparatus, work could be developed. Nevertheless, it seems to the writer that it is not entirely fair to charge any apparatus with failure to accomplish that which it is incapable of doing. The fact that available work is lost in the boiler is no fault of the boiler, but results from the deliberate choice of the plant designer. He does not choose to introduce apparatus to develop work at the expense of cooling the hot gas rising from the furnace. He does not attempt to produce a high temperature as such. In fact, if satisfactorily complete combustion could be attained by some process at lower temperatures, many boiler-room difficulties would be reduced or eliminated. The fact that the temperature is high, which gives rise to a large

[1] Technical Engineer of Power Plants, The Detroit Edison Company, Detroit, Mich. Mem. A.S.M.E.

proportion of available work, is an unfortunate consequence of the only methods now known for securing the release of the desired heat from fuel.

Inasmuch as most power-plant apparatus over which the operator has any significant measure of control is largely heat-transfer equipment, it seems to the writer that the first-law heat balance, already so familiar, is preferable to the proposed second-law statement, which, while admittedly better for the turbine, appears to be less satisfactory for everything else in the plant.

J. G. BLUNT.[1] The author has extended the second-law analysis (which is usually employed to measure the efficiency of the use of steam in the engine) to the power plant as a whole, from the furnace to the final exhaust. This form of analysis is superior to the first-law analysis in that it separates the non-available portion of heat energy in the various mediums from the portion that would be available if the ideal Carnot cycle of maximum efficiency could be realized in practice. It gives a much clearer idea of the relative importance of the losses at different stages of the process of conversion of the heat energy in the coal into mechanical energy. Except in the case of the steam, however, the basis would seem to be somewhat arbitrary, in that the mediums other than the steam, do not, in the actual process, give up their energy by a cyclic operation in the engine itself. The writer believes that this form of analysis constitutes a distinct advance.

THE AUTHOR. I wish to thank Professor Butterfield for checking the calculations in this paper. The small differences are due to the use of Goodenough's steam tables in the original calculations, while Professor Butterfield used the Marks and Davis tables. His explanation of the discrepancy noted in the original test data as being due to an inaccuracy in the measurement of the indicated horsepower may be correct. However, this discrepancy was not attributed to an error in the measurement of the temperature of the exhaust steam, but of the superheat in the steam initially before it entered the engine cylinders.

The use of the absolute temperature in the formula for the specific heat of furnace gases given in Par. 10 is not an error, as this was done deliberately. The absolute temperature is more convenient to use in the various calculations indicated than degrees on the fahrenheit scale. The coefficient given expresses the variation of the specific heat with absolute temperature as closely as the specific heat is known for the mixtures of furnace and stack gases.

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The list indicating the Detailed Distribution of Losses of Available Heat Energy is not correct. Item 3, the loss in available energy "in transformation of chemical energy to sensible heat of the furnace gases" must include the "unavailable heat carried away by the stack gases" as listed under item 6. Item 9 should not appear in this list because the "unavailable heat in the exhaust steam" is not a loss of available energy. The available energy lost by imperfections in the engine cylinders does not appear in this list.

Table 1 of Professor Butterfield's discussion should not be headed Summary of Losses of Available Energy since it includes item 10, which is not a loss. The items of Table 1 do not correspond to those in the list previously given. Thus item 9 in Table 1 is the available energy lost in the exhaust steam while it is previously listed as "unavailable energy in the exhaust steam." Item 8 in Table 1 is the loss of available energy due to imperfections in the engine cylinders, which is not mentioned in the preceding list, item 8 there being given as "available heat in the exhaust steam." Item 6 in Table 1 is the available energy lost in the stack gases, while this item is previously stated to be the "unavailable heat carried away by the stack gases." Item 5 in Table 1 is the unavailable energy in the stack gases, this item being previously listed as the "loss of available heat during cooling of the stack gases." Incidentally, the loss of available energy in the stack gases is due to the fact that they are not cooled. The sum of items 3 and 5 in Table 1 is equal to item 3 as worded in the detailed distribution of losses.

No. 1963

COMPARISON OF ACTUAL PERFORMANCE AND THEORETICAL POSSIBILITIES OF THE LAKESIDE STATION

By M. K. Drewry,[1] Milwaukee, Wis.

Junior Member of the Society

To show the degree of perfection in the design and operation of a high-efficiency, straight-Rankine-cycle station, pointing out the magnitude of each individual heat and availability loss occurring in actual practice, is the object of this paper. Operating and test figures of the Lakeside station have been assembled and a comparison of theory and practice has been recorded.

FOR THE designer of a power station to predict accurately the efficiency of the finished plant is obviously quite more involved than to ascertain the economy of the theoretical cycle which the station employs. The ever-present and elusive difference between theory and practice is the unknown with which he must contend. To record and explain the extent and cause of the difference in one particular example, that of the Lakeside station, is the primary purpose of this paper. A discussion of how the data with which the comparison has been made were collected, and how operating practices at the station affect the economy, is given also.

Method

2 Comparison of theoretical possibilities and actual performance has been made during a given period of best economy, October 10 to November 10, 1924, by the following means:

1 Computing plant "efficiency ratio" when the station is most economically loaded, estimating 2 per cent radiation and miscellaneous losses

2 Finding losses attendant with actual operation, due to uneconomical loading of turbines and boilers and to starting and banking, etc., using actual operating data

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3 Calculating the theoretical cycle efficiency with which a perfect station would have worked, and multiplying this efficiency by the efficiency ratio of the actual station

4 Equating actual and calculated heat consumptions to check allowances made in the calculations for radiation and miscellaneous losses.

Results

3 That the actual operating heat consumption of the station is within 5 per cent of that theoretical standard which considers all equipment operating most economically, is shown by Table 1. The basis for each of the figures used is discussed in subsequent paragraphs. The operating data are given in Table 6.

TABLE 1 COMPARISON OF THEORETICAL POSSIBILITIES AND ACTUAL PERFORMANCE

	Per cent
Most Economical Loading	
1 Boiler efficiency (computed from losses)	91.6
2 Turbo-generator efficiency ratio (from tests)	76.2
3 Consumption of auxiliaries	
Electrical auxiliaries (from meters)	4.25
Steam (credit given for heat return to condensate; taken from operating data)	1.84
4 Coal drying (from operating data)	0.25
5 Radiation and miscellaneous losses (by difference to equalize items 14 and 15)	2.2
6 Efficiency ratio,[1] most economical loading	64.0
Losses Attendant with Actual Operation (Calculated from Operating Data)	
7 Boiler room	
Uneconomical loading of boilers	0.25
Banking and starting cold boilers	1.98
8 Turbine room	
Uneconomical loading	1.75
Starting	0.26
9 Unusual operations	
Entire plant	0.22
10 Total operating losses	4.49
11 Calculated efficiency ratio, average operating conditions	61.1
12 Theoretical cycle efficiency (Marks and Davis tables)	35.4
13 Calculated plant thermal efficiency	21.6
	B t.u. per kw-hr.
14 Corresponding calculated heat consumption	15,800
15 Actual plant heat consumption	15,800

[1] A measure of equipment efficiency. Actual boilers, turbines, and auxiliaries have a combined efficiency of 64 per cent that of perfect equipment.

4 The calculations of Table 1 show that less than 5 per cent (see item 10) need be added to the best theoretical performance of the station to obtain its heat consumption under actual operating conditions. In terms recently introduced, its "operating efficiency ratio" is approximately 95 per cent.

5 Of the 5 per cent difference, about one-quarter of one per cent can be attributed to uneconomical boiler loading, 2 per cent to banking and starting cold boilers, 2 per cent to uneconomical loading and starting of turbines, and the remainder to unusual operations and miscellaneous losses. The theoretical-plant-efficiency

calculation allows 2.2 per cent for radiation and miscellaneous losses, a large share of which are constant regardless of load carried.

Basis of Calculations

6 *Boiler Efficiency.* Continuous measurement of the most significant measurable losses in boiler operation allows boiler efficiencies to be quite accurately deduced by totalizing these comparatively small losses instead of equating the large inputs and outputs as done in complete boiler testing. Reliance is placed upon the method since it indicates the same efficiencies which extensive monthly calculations have shown must be realized in actual operation.

7 Table 2 is a heat balance calculated from data gathered during the period in question.

TABLE 2 HEAT BALANCE

	B.t.u.	Per cent
Heat per lb. of coal	13,691	100
Heat loss due to moisture in coal	26	0.19
Heat loss due to water from combustion of hydrogen	487	3.55
Heat loss due to moisture in air	10	0.07
Heat loss in dry chimney gases	453	3.31
Heat loss due to incomplete combustion of carbon	68	0.5
Heat loss due to unconsumed hydrogen, hydrocarbons	0	0
Heat radiated but not reclaimed; and unaccounted-for losses	103	0.75
Heat absorbed by boiler, economizer, and superheater	12,544	91.6

8 Of the above losses, that of combustible in the flue gases is the only one not measured. No attempts to learn the percentage of combustible in the flue gases have as yet been successful, for the particles of ash have been found so extremely fine that no convenient filter will remove proportional quantities of ash and coal. The tan color of the solids emerging from the stacks during normal operation appears no darker than that of pure, calcined ash, but inability to measure the carbon losses during periods of starting, when the carbon losses are of some consequences, has required an estimate of 0.5 per cent to be used.

9 Loss due to unconsumed hydrocarbons cannot be detected in the furnaces until excess air is decreased below 15 per cent, and then CO appears in small quantities. Since the average operating air supply is 22 per cent in excess of the theoretical requirement and is maintained closely, the losses due to unburned hydrocarbons are negligible.

10 Radiation losses are undoubtedly small. Furnace-casing temperatures average about 110 deg. fahr., and practically all heat losses contribute, with the supply of heat in the generator cooling air, to maintain the combustion-air temperature at 100 deg. fahr. The coolest daily average combustion-air temperature recorded the past winter was 80 deg. fahr. Previous tests on units in the first boiler installation placed radiation and unaccounted-for losses at 2 per cent, but the newer boilers, which carried the largest

percentage of the load during the period considered, have considerably lower radiation losses.

11 *Turbo-Generator Efficiency.* This is a weighted value derived from test results averaging the Rankine-cycle efficiency ratio (Marks and Davis steam tables) of the four units installed and operated during the time considered. As previously inferred, no steam-extraction operation occurred during the period under consideration. The efficiency ratio recorded includes generator losses and excitation energy.

TABLE 3 DIVISION OF ENERGY TO AUXILIARIES

A—Electrical Auxiliaries	Per cent
1 Station lighting	0.1
2 Coal handling:	
Car dumper	0.0006
Magnetic pulley	0.006
Crushers	0.02
3 Preparation house:	
Conveying green coal	0.003
Pulverizing	1.08
Conveying pulverized coal	0.025
Total	1.110
4 Boiler room:[1]	
Induced-draft fans	0.50
Coal-feeder motors	0.08
Primary air fans	0.25
Boiler-feed pumps	0.62
Air compressor, machine shop, etc.	0.25
Total	1.65
5 Turbine room:	
Circulating pumps	0.88
Hotwell pumps	0.29
Air removal (on one unit)	0.17
House service and miscellaneous	0.03
Total	1.37
Total energy for electrical auxiliaries	4.25
B—Steam-Driven Auxiliaries	
1 Boiler room:	
Exhaust heat reclaimed:	
Boiler-feed pumps	1.24
Induced-draft fans (one boiler)	0.02
House service pump	0.01
Exhaust lost:	
Steam jets on burners	0.01
Boiler soot blowers	0.2
Economizer soot blowers	0.43
Superheater soot blowers	0.31
Ash conveyors	0.4
2 Turbine room:	
Steam-jet air pumps (heat reclaimed)	0.36

[1] Estimated from tests.

12 *Consumption of Auxiliaries.* The consumption of all electrical auxiliaries is measured with integrating meters, and that of steam auxiliaries is obtained by recording the time operated and multiplying by factors learned by test of each unit.

13 Credit for heat returned to the condensate by exhaust steam from the auxiliaries has been allowed. Table 3 shows the itemized auxiliary consumptions.

14 *Drying of Coal.* The boiler efficiency recorded in the previous tabulations does not consider coal-drying charges, though the usual monthly efficiencies tabulated elsewhere do. Coal for drying is estimated volumetrically, and is probably accurate to within 5 per cent.

15 Lack of daily coal temperatures does not allow computation of drier efficiency. The fuel requirements vary from summer to winter to the extent of 3 lb. of coal used per ton dried per 1 per cent of moisture removed. The average summer and winter values are about 2 and 5 lb., respectively. Table 4 gives the average operating figures for the period considered.

TABLE 4 COAL CONSUMPTION OF COAL DRIERS

Total coal dried, tons	22,818
Coal used for drying, tons	67
Moisture of coal, drier inlet, per cent	4.54
Moisture of coal, drier outlet, per cent	2.82
Moisture removed, per cent	1.72
Pounds of coal used per ton dried per 1 per cent of moisture removed	2.90

16 *Radiation and Miscellaneous Losses.* This value was first set by estimate at 2 per cent, but was later changed to 2.2 per cent to cause agreement between calculated and actual results.

17 Chief of the unreclaimed radiation losses is the 25-deg. drop in superheated-steam temperature from boiler room to turbine room. Blow-down losses are slight, 6 B.t.u. per kw-hr., the character of Lake Michigan water being such that the few dissolved solids settle out at the blow-off pads, and that the concentration of the boiler water is caused to remain continuously very low, less than 0.1 deg. B.

18 Another source of loss is that of steam leakage through radiant superheater elements which were then in the development stage. The period of best economy chosen for this analysis occurred at a time when station make-up was lowest, indicating lowest superheater leakage. An accurate estimate of this loss based upon the quantity of make-up places it at 0.9 per cent, for the theoretical make-up required is 2.2 per cent and the actual average was 3.1 per cent. Though accuracy in make-up measurement is not in itself of a high degree, when applied as a percentage of the total station water any errors in the commercial water meter appear slight. The theoretical make-up of 2.2 per cent has been reached since steam leakage has been stopped by the use of different material composing the elements. The 0.9 per cent correction was made directly upon station heat consumption, and is not entered under "miscellaneous losses" since obviously the experimental work cannot be charged against the efficiency of the station.

19 Pop-off losses must obviously be small since they are included in the 2.2 per cent radiation and miscellaneous losses.

20 *Loading of Boilers.* Figs. 1 and 2 show the average percentage of time during which various ratings are carried, and the

expected efficiency with which each of the ratings are carried. Fig. 1 was plotted from actual operating data of November, 1924:

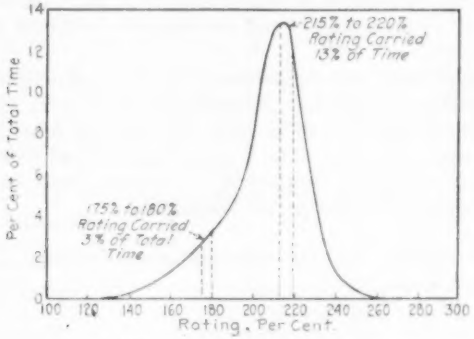


FIG. 1 BOILER LOADING

Showing percentage of total time that various percentages of rating are carried. Calculated in steps of 5 per cent of rating. (See notes on curve.)

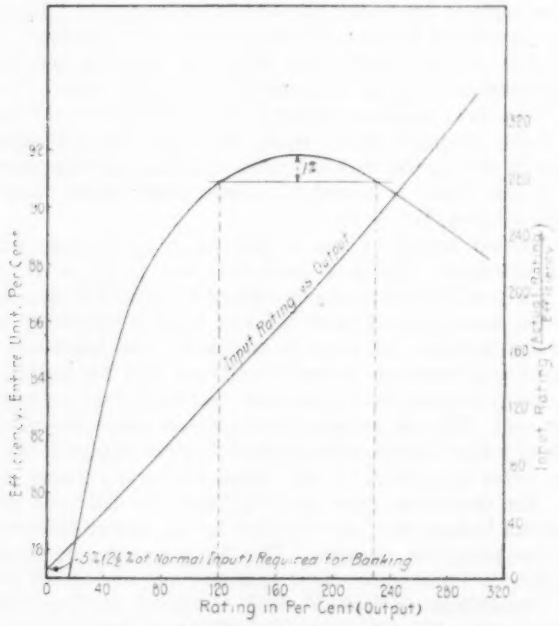


FIG. 2 ESTIMATED BOILER EFFICIENCY

Based upon measured losses observed during actual operation with test instruments. Banking requirements measured by computing from feeder screw speeds the coal feed necessary to maintain full boiler pressure.

Fig. 2 has characteristics derived from observing losses at various percentages of rating. The slope of this boiler-efficiency curve

agrees with the manufacturer's estimate, and the curve will probably be closely paralleled by test results this coming summer. That the efficiency curve is so " flat " that only 0.25 per cent loss due to uneconomical boiler loading occurs is shown in Table 1.

21 Tests have indicated 2½ per cent of normal coal feed (at 215 per cent of rating) necessary to bank a boiler and maintain its pressure, and operating data reduce the amount below 2 per cent because the proportion of banked and inactive hours to active hours is less than unity. To retain heat in a boiler, all air passages to the furnace are closed immediately upon stopping coal feed, and the unit is practically " bottled " tight so that convection losses are a minimum.

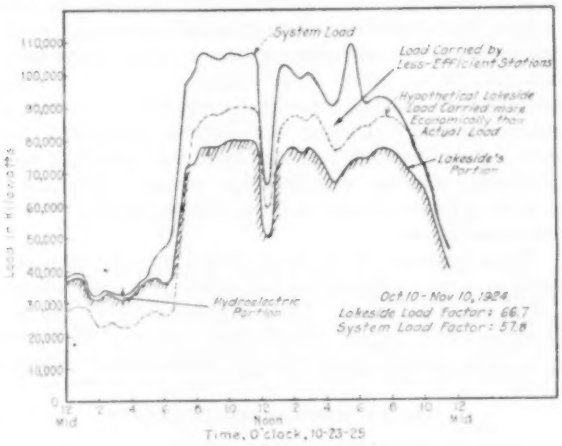


FIG. 3 LAKESIDE LOAD CURVE

System and Lakeside loads on an average day (October 23, 1924). Lakeside load factor from October 10 to November 10 was 66.7; system load factor during same period was 57.8. The dotted lines show a hypothetical load which the station could have carried ½ per cent more efficiently.

22 *Turbine Loading.* Under " uneconomical-loading losses " have been set the weighted losses due to increase of water rate above or below that at the most economical load. It is the practice at the Lakeside station to keep all units, whenever possible, accurately at their most economical loads, these loads being approximately 95 per cent of the full rated generator capacity in the case of the 30,000-kw. units. The load curve illustrated in Fig. 3 shows the greatest throttling loss to occur during the early morning period, 12 to 6 a.m., when approximately 35,000 kw. load was carried by 50,000 kw. capacity. Lakeside's portion of the system load during the period under consideration was limited by a shortage of transmission-line capacity.

23 The question of load factor influencing economy enters in this connection. That the ratio of average load to maximum

demand load is of smaller consequence in the economy of Lakeside than is the matter of turbine loading is proved by a short calculation. If the midnight-to-morning load is decreased 10,000 kw. and the day load is increased the same amount (see dotted curve in Fig. 3), then banking losses would be increased a maximum of ¼ per cent. Lowering the early morning load for six hours would allow operation of a single 30,000-kw. unit (see Fig. 3), effecting a gain of 0.4 per cent on the day's output. The average full load of 75,000 to 80,000 kw. shown in Fig. 3 was unfortunately above the most economical capacity of the combination of units used (two 30,000-kw. units and one 20,000-kw. unit), and if increased to 90,000 kw., warranting the starting of another unit, it would have netted an additional gain of 0.3 per cent in the turbine room.

24 The comparative greater sustained economy with which pulverized-fuel boilers can carry widely variable loads than steam turbines can carry given loads explains why load factor has not as important an influence upon economy as has the question of turbine loading. During the period considered the load factor of Lakeside equipment was high, but turbine loading was uneconomical, causing compensatory effects. Economy results obtained after Lakeside carried the entire system load corroborated the above deductions.

25 Three-quarters of an hour steam consumption at a rate measured by tests is allowed for starting the turbo-generator units.

26 *Unusual Operations.* Table 5 gives a list of the entries in the October, 1924, bonus calculations, showing the nature and extent of these miscellaneous charges upon station economy.

TABLE 5 MISCELLANEOUS CHARGES

	Pounds of steam
Testing emergency governors on Nos. 1 and 3 turbines	110,000
Testing emergency governors on Nos. 1, 2, 3, and 4 circulating pumps	20,200
Cutting in Nos. 2 and 3 headers	12,000
Siphons on smoke washer	62,275
Setting safety valves, No. 2 boiler room	6,630
Equivalent steam for refuse taken from mill rings	84,219
Equivalent steam for weight of iron taken from magnetic pulley	36,805
Equivalent steam for coal taken for drying out new furnaces under construction	158,674
Equivalent steam for boiler fillings and special blow-downs	452,955
Total	943,758
Percentage of plant output	0.22

27 *Actual Plant Economy.* This figure is obtained twice monthly, and the one recorded in item 15 of Table 1 is an average of two biweekly accountings. The plant heat consumption for the first two weeks considered, October 10 to 26, 1924, was 15,869 B.t.u. per kw-hr., and that for the remainder of the monthly period was 15,721, making an average of 15,795 B.t.u. per kw-hr.

28 Coal is weighed on government-sealed track scales just previous to its entry into the plant grounds proper. These coal weights are checked with those of the coal dealer before payment is made.

29 Sampling is done in the crusher house as the crushed coal leaves the hammer mill and starts upward on the conveyor belt to the preparation house. When coal is being conveyed, 25 samples of the thoroughly mixed coal are taken from the belt, each at 10-minute intervals, until the accumulated pile of 35 lb. represents 500 tons of coal. This sample coal, with the exception of about 5 lb. for moisture determination, is then crushed finely in a test crusher which rejects nine-tenths of the prepared sample and retains one-tenth. From this smaller portion is taken the sample from which the heat content is determined. All sampling is done by regular operators.

30 Heat content is determined by bomb calorimeter in the company's laboratory. Duplicate samples sent to the Bureau of Mines have checked within an average of 25 B.t.u. per kw-hr. The average proximate analysis of coal during the period considered is as follows:

	Per Cent
Volatile matter	31.4
Fixed carbon	59.4
Moisture	4.02
Ash	9.2
Sulphur	1.1
B.t.u. per lb. as received	13,141
B.t.u. per lb. dry	13,691

31 The lack of frequent ultimate analyses lends some doubt to the boiler-efficiency calculations in which the losses due to hydrogen burning to water are computed. The analyses possessed and the average analyses made by the Bureau of Mines of coal from the communities in which Lakeside coal is mined average as follows:

	Per cent
Sulphur	1.1
Hydrogen	4.95
Carbon	78.0
Oxygen	6.6

32 The maximum probable error in the recorded hydrogen content is thought to have a maximum of 0.25 per cent effect upon the boiler-efficiency calculations.

33 The average fineness to which coal was pulverized during the October 10 to November 10 period was as follows:

Mesh	200	100	50	40	30	20
Per cent through	64	81	96	99.2	99.8	100

34 The average moisture content at the boiler bins was 1.77, showing that in being pulverized about 1 per cent of moisture was removed.

35 Coal samples are also continuously taken from the pulverized-fuel bins, but are used merely to determine final moisture content and fineness. Boiler efficiencies and plant heat consump-

tion submitted make no deductions for the coal used in drying, but all computations consider all of the coal weighed into the station reaching the boiler.

36 To determine the amount of coal on hand when the biweekly "coal check" is made, all storage bins are previously emptied so that only the coal in the boiler bins and in the small coal-drier bins need be estimated. Errors in measuring the coal in the active bins are negligible when charged against two weeks' operation. On one occasion two independent parties separately estimated the coal in storage, and in support of reasonable deductions made previous to the check, the results were as follows:

Party	No. 1	No. 2
Estimate of pulverized-coal bins, tons	387	382
Discrepancy, per cent	1.5	
Per cent of coal in pulverized-coal bins to total consumed	1.45	
Resulting variation in computation of plant heat consumption, per cent	0.02	

37 Since the amount of coal estimated during a monthly check is only 1.45 per cent of the total consumed, a 1.5 per cent possible discrepancy in estimating that relatively small amount of coal causes an entirely insignificant difference in calculated results.

38 At the same time that coal estimates are made, electrical readings are taken of the output of each unit during the biweekly period. These are later checked for gross reading errors by comparison with the readings that are taken at midnight every 24 hours.

39 Accurate checks of the integrating meters and instrument transformers made shortly after November 10 showed that the following corrections should be applied on the output of each unit:

Unit No.	1	2	3	4
Correction, per cent	−0.63	−0.68	−0.60	−1.48

40 The average correction, weighted in proportion to the energy generated by each unit, was 0.95 per cent. This factor has been applied in the recorded station heat consumption.

41 No attempt is made to obtain economy data more frequently than twice each month, for it is felt that the element of inaccuracy and doubt attendant with more frequent readings renders such data of no value. Rather than rely upon such data, an immediate check on possible losses is furnished by 68 recorder charts and many indicated readings recorded by the operators and gathered at midnight of each day. These charts are in turn examined by the boiler-room engineer, by the first assistant engineer, and by the engineer in charge, and are then immediately averaged by planimeter and the averages given to the operators. A weekly summary is made and distributed in such numbers that every operator can see and study a copy without leaving his work.

Bonus System

42 To encourage economy in the use of coal, steam, and electricity, all operators at the station are paid a percentage of the saving they effect in excess of a predetermined standard.

43 Obviously, to be effective the bonus system needs to be based upon accurate data, therefore, only monthly economy figures are used. Calculation of the standard station efficiency is made each month, and all operators are paid 45 per cent of the actual coal saved above the standard set. The bonus paid has varied from 8 to 15 per cent of the stipulated wage since the bonus system has been inaugurated.

44 Accuracy in the data gathered and of the computations made is realized to be of much importance when one learns that because of the comparative smallness of the force employed and of the magnitude of possible money savings a 1 per cent additional saving in plant economy will net each employee about 4 per cent additional bonus. Because of this rapidly multiplying "lever," a close check of the conditions of all plant equipment must be kept, and a test force is maintained first to determine proper standards and then to detect any later change in the efficiency of equipment.

Minimizing of Plant Losses

45 Keeping constant watch of plant equipment and data suggests means of decreasing losses. Better performance of one unit of equipment than of other similar units prompts improvement of the poorer units, and thus best possible results are obtained from every unit.

46 No standards of air leakage into condensers were determined until test equipment showed one unit leaking less than 5 cu. ft. per min., the equivalent of the leakage through a $\frac{1}{8}$-in. orifice. It was reasoned that the other 30,000-kw. units could be made equally tight, and at present the air leakage of all units averages about 5 cu. ft. per min. The exhaust pressures maintained in average operation are such that the terminal temperature difference of the exhaust-steam and circulating-water outlet temperatures vary from 11 deg. with 40-deg. inlet water to 8 deg. with 70-deg. inlet water. Tables showing the expected exhaust pressures at all loads and circulating-water temperatures are used by the operators to detect any slight decrease in vacuum.

47 Turbine loading, unlike boiler loading, is given careful attention. Except during peak- and low-load periods, the 30,000-kw. units are kept accurately at their most economical loads by making the overload valve gear inactive. A special handwheel is used for this purpose. In such cases the pressure drop through the primary valve is reduced to 7 lb., and is maintained at that point at all times irrespective of slight changes in frequency.

Before studying turbine loading, in many instances units were simultaneously operating above and below their individual most economical loads, with losses indicated by Fig. 4.

Boiler-Room Losses

48 Maintenance of maximum boiler and economizer air leakage at or below 1.5 per cent CO_2 decrease in the gases has been set as a standard after weekly observation of this source of loss. The CO_2 measurements are taken at three points above the first bank of tubes and at each of the fan discharges where thorough mixture of air and gases is thought probable. This 1.5 per cent air leakage includes infiltration through the economizer bottom, a difficult section to keep tight because of water treatment when washing

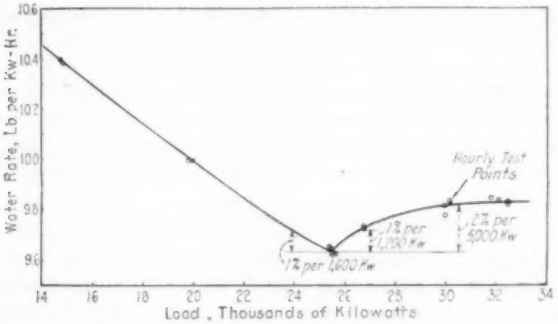


Fig. 4 Turbine-Water-Rate Variation

Tests of No. 3 unit, Lakeside, showing the rapid change of water rate at the most economical load, explain why turbine loading may have greater effect upon plant economy than load factor, in view of the low banking losses of the pulverized-fuel boilers.

economizer tubes, though this latter infiltration causes no appreciable thermal losses.

49 Because of the flatness of the pulverized-fuel boiler-efficiency curve obtained in actual operation and of the small banking losses, an exhaustive study of various loading combinations has shown little difference in the resulting economy. Maintenance and other considerations determine ratings of individual boilers rather than does economy.

50 Table 6 gives operating data collected during the period considered and the months following. Averages were taken of operation during the five full working days of each week and when boilers were operating at 200 per cent rating or over.

51 Overall station performance during the same time was as given in Table 7.

Conclusions

52 A modern power station may be so designed and operated that its ultimate efficiency will be within 5 per cent of the combined

efficiency of its component equipment working at most economical load. In other words, a 5 per cent allowance can represent all

TABLE 6 OPERATING DATA, LAKESIDE PLANT

Plant	Oct. 10 to Nov. 10	Dec.	Jan.	Feb.	March
1 Daily coal consumption, tons [1]	867	915	934	825	966
2 Daily output, million kw-hr	1.423	1.528	1.557	1.542	1.629
3 Make-up water, per cent	3.12	3.80	3.19	2.85	2.51
Boiler Room					
4 Rating, per cent	213	227	228	228	225
5 Boiler hours, daily	141.7	136	138.5	136.94	142.4
6 Banked hours, daily	17	18 1	19.5	19.3	28.63
7 Combustion-air temperature, deg. fahr.	99	82	84	84	83
8 CO_2, per cent	14.7	14.8	14.8	14.8	14.7
9 Steam temperature leaving superheater, deg. fahr.	702	687	710	721	717
Gas Temperatures, Deg. Fahr.					
10 Economizer inlet	564	572	573	574	568
11 Economizer outlet	232	237	243	244	248
12 Drop in temperature	332	335	330	330	320
Feedwater Temperatures, Deg. Fahr.					
13 Economizer outlet	251	248	254	258	260
14 Economizer inlet	119	121	133	137	137
15 Rise in temperature	132	127	121	121	123
Turbine Room					
Temperatures, Deg. Fahr.					
16 Throttle, Nos. 1 and 2 units	671	647	640	....	619
17 Throttle, No. 3 unit	682	660	677	683	684
18 Throttle, No. 4 unit	689	658	688	701	695
19 Throttle, No. 5 unit	....	673	693	702	701
20 Condenser hotwell	73	63	64	63.2	61.8
21 Circulating-water outlet	67	53	56	55.9	54.4
22 Circulating-water inlet	53	38	41	41.1	40.7
23 Circulating-water rise	14	15	15	14.6	13.7
Pressures, Lb. Gage and In. Hg Abs.					
24 Throttle, all turbines	288	286	289	291	293
25 Exhaust Nos. 1 and 2 units	0.91	0.63	0.56	....	0.56
26 Exhaust Nos. 3 and 4 units	0.94	0.78	0.76	0.76	0.74
Preparation House					
27 Per cent moisture, drier inlet	4.54	5.58	6.75	7.64	7.18
28 Per cent moisture, drier outlet	2.82	2.52	2.48	2.85	2.72
29 Per cent moisture removed	1.72	3.06	4.27	4.79	4.46
30 Tons coal used in drier daily	1.84	6.08	6.96	7.56	8.23
31 Lb. coal used per per cent moisture removed per ton dried	2.90	4.72	3.53	3.41	3.82

[1] Estimated from electrical output, using previous month's B.t.u. per kw-hr.

TABLE 7 BOILER AND STATION PERFORMANCE DURING THE SIX MONTHS ENDING APRIL 1, 1925

Month	Plant heat consumption, B.t.u. per kw-hr.	Boiler efficiency, steaming only, per cent
October, 1924	16,000	89.71
November, 1924	15,945	91.34
December, 1924	16,308	89.67
January, 1925	16,066	88.85
February, 1925	15,861	90.05
March, 1925	15,930	[1] 89.7

[1] Reduced due to greater use of oldest equipment.

losses attendant with actual production of power, such as banking of boilers, starting of turbines, uneconomical loading of turbines, etc.

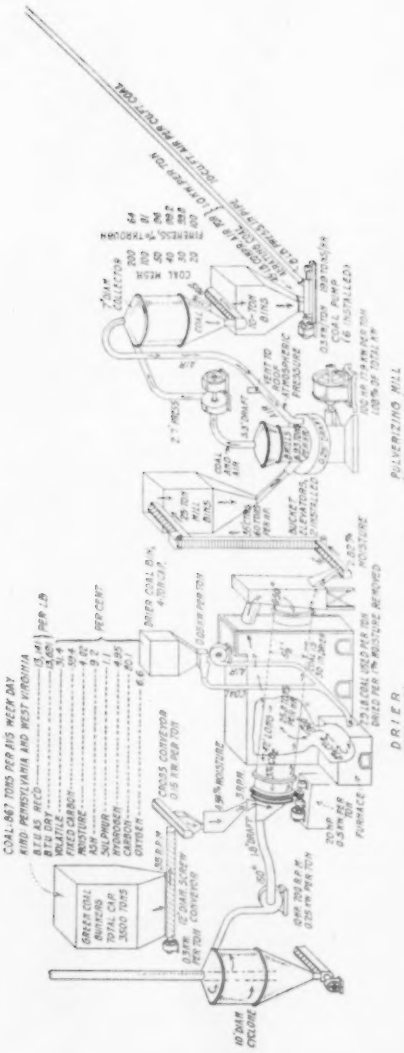


FIG. 5 OPERATING DATA AND DESIGN FIGURES OF THE LAKESIDE STATION

(For boiler and turbine rooms see following page.)

Showing average temperatures, pressures, rates, etc., occurring during the period of October 10 to November 10, the economy data of which are analyzed in the paper. Particular attention is called to the percentage energy consumption placed on or near each unit of equipment, showing its relative consumption. Design data of sizes, ratings, etc., have been added wherever convenient.

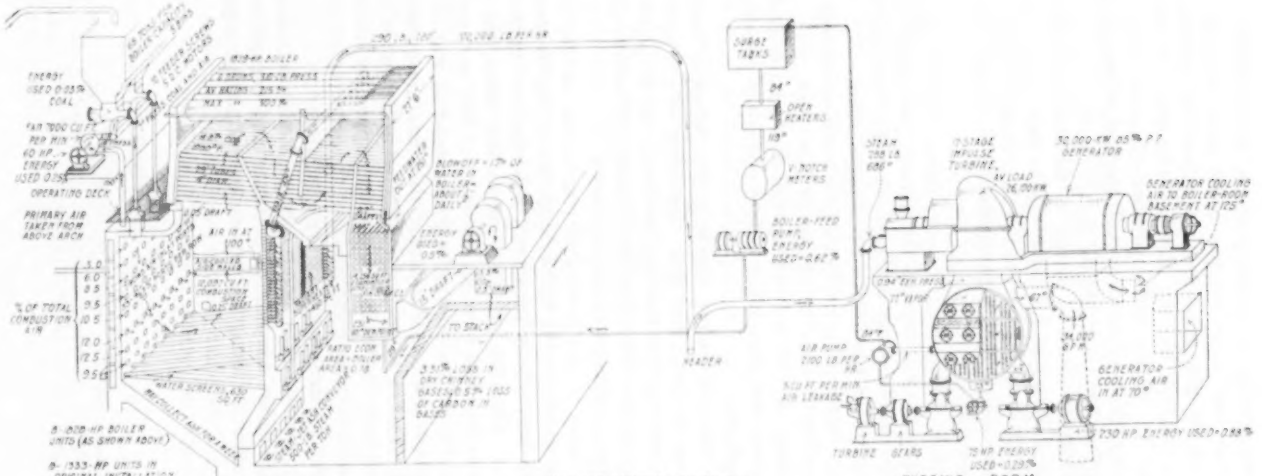

BOILER ROOM
EFFICIENCY OF UNITS = 91.6%
TOTAL ELECTRICAL ENERGY USED=1.65%
COMBINED MAXIMUM EFFICIENCY OF BOILER-ROOM AND TURBINE-ROOM EQUIPMENT WAS 65% AT MOST ECONOMICAL LOADING, 62% WHEN CARRYING ALL LOADS
TURBINE ROOM
EFFICIENCY RATIO OF MAIN UNITS = 76.2%
TOTAL ELECTRICAL ENERGY USED= 1.3%

FIG. 5 (Continued)

53 The combination of boilers, turbines, and auxiliaries, allowing for radiation and miscellaneous losses, may work at an efficiency ratio of 64 per cent in a pulverized-fuel station. This percentage multiplied by the thermal efficiency of the cycle employed and by 95 per cent operating-efficiency ratio should give the approximate ultimate overall thermal efficiency of any pulverized-fuel station using the usual equipment.

54 Totalizing the losses of energy in the Lakeside station, all of which except those of radiation and a few minor ones are measured, and applying the total to the theoretical efficiency of the cycle employed, shows that a performance of 15,800 B.t.u. per kw-hr. is reasonable.

55 Decrease in boiler-unit efficiency below that at most economical rating due to operation throughout the range of loads found in actual practice may aggregate less than one-half of one per cent in the case of a pulverized-fuel installation.

56 Banking losses of a pulverized-fuel station operating at 65 per cent load factor may be limited to 2 per cent.

57 The energy cost of pulverizing coal was 1.1 per cent, and that of conveying and pumping it before and after pulverization was almost negligible.

58 Drying the coal before reaching the furnaces consumed 0.25 per cent of the total coal flow. Assuming 30 per cent drier efficiency and allowing for the heat necessary to evaporate this moisture in the furnace if not previously dried, the net energy cost of the coal-drying process was 0.16 per cent.

59 Of the losses in the process of combustion in the furnace and of heat absorption in the boiler, superheater, and economizer, 45 per cent are unavoidable because of hydrogen and moisture in the coal and air. Allowing for these unavoidable losses, the boiler unit may be considered to have an efficiency of 95.4 per cent.

60 Load factor in the case of a pulverized-fuel station may have less influence upon coal economy than has the question of turbine loading. During the period considered in the paper, Lakeside had a high load factor, but was handicapped by having to carry loads of such certain magnitude that the economy of turbine loading was poorer than under average conditions.

61 The employment of a bonus system, besides encouraging economy, demands the keeping of accurate and comprehensive records, which aid in bettering the performance of plant equipment.

No. 1964

RECENT INVESTIGATIONS IN TURNING AND PLANING AND A NEW FORM OF CUTTING TOOL

By Dr. Hans Klopstock,[1] Berlin, Germany
Non-Member

This paper gives an account of experiments conducted in the machine-tool laboratory of the Polytechnic Institute of Berlin. The cutting tests proper were preceded by a study of the lathe and the instruments used, to ascertain their behavior under various conditions. In the main tests it was aimed to determine the influence upon the cutting forces of the following factors: Cutting speed, chip section, shape of tool, and characteristics of the materials cut.

As a result of these observations a new form of tool has been designed which, the author claims, permits a higher cutting speed and a larger chip section than is practicable with the standard tools now used. The results of some of the tests made in various German railway shops are given, in which the relative life of the cutting edges of the new and the old tools under exactly the same conditions was compared.

It is claimed, from results obtained in various foreign shops, that on the average production can be increased about 30 per cent by means of the new tool.

WITH increasing wages, the problem of efficient tools and working methods becomes of ever-increasing importance. Scientific investigations covering turning and planing which the author has carried on during a period of two years in the machine-tool laboratory of the Polytechnic Institute of Berlin, have resulted in certain improvements which have shown themselves to be of material economic value.

2 It was believed from the start that a study of this character could lay claim to scientific accuracy and absolute dependability only if every mechanical measurement made could, in some manner, instantly be checked. The only manner in which this seemed

[1] Pres., Klopstock Steel Co.

Translated by L. C. Marburg, Mem. A.S.M.E., contributed by the Machine Shop Practice Division, and presented at the Spring Meeting, Milwaukee, Wis., May 18 to 21, 1925, of The American Society of Mechanical Engineers.

possible was by measurement of the electric power input into the driving motors. A check of this character, however, required a complete knowledge of all the characteristics and losses of the machine tool used. The cutting tests were therefore preceded by

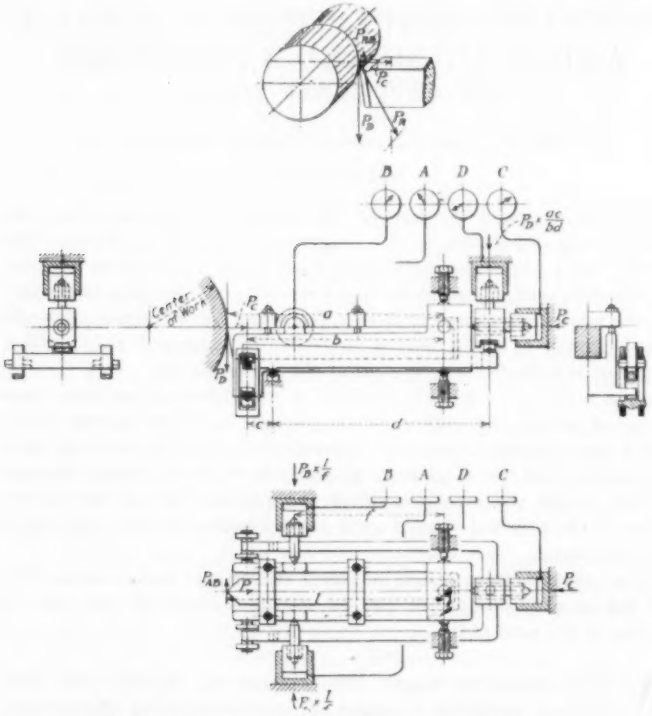


FIG. 1 DIAGRAM OF TOOL SUPPORT AND GAGES

a most thorough study of the lathe on which the tests were conducted; and incidentally, of course, all electric meters, gages, and other measuring devices used were checked as to their behavior under all possible conditions.

INVESTIGATION OF LATHE AND INSTRUMENTS

3 *Test Equipment.* The lathe used in these tests has a height of centers of about 18 in. and a distance between centers of about 118 in. It has five different spindle speeds and in addition the speed of the 50-hp. driving motor can be varied from 350 to 1000 r.p.m. The speed of the spindle can thus be changed, in small steps, from 7 to 820 r.p.m. The feed gear has eight speeds, which

will permit feeds within the limit of 0.0037 in. and 0.1 in. per revolution. In the tests it was driven by a separate 5-hp. motor.

4 The tool support used in the tests was of a type which divides the total pressure on the tool into three components, a vertical component D, a surfacing component C, and a traversing component B. The tools were always accurately set at the height of the centers, in which position all measurements taken checked within $\pm\frac{1}{2}$ per cent. Fig. 1 shows this support and its gages

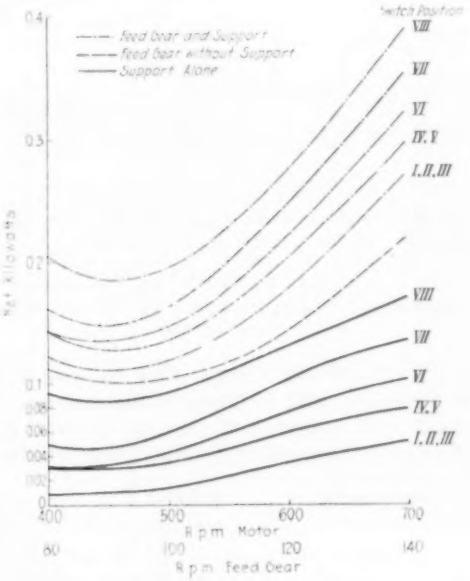


FIG. 2 POWER CONSUMPTION OF FEED GEAR RUNNING IDLE

diagrammatically. A recording gage was also connected which could record all three components at the same time; it was interconnected with a recording wattmeter showing motor input in such a manner that both could be started and stopped by the same operation, so that a perfect check on readings could be had.

5 A detailed report of all tests and calibrations of tool-support instruments, gages, and motors is unnecessary here. It is sufficient to say that they were conducted in the most painstaking manner and carried to a point where all possible errors were eliminated.

6 *Power Consumption of Feed Gear.* The power consumption of the feed gear, running idle, was determined at each gear ratio and with different motor speeds. It proved extremely small as compared with the power requirements of the main spindle, as was to be expected. The curves in Fig. 2 show the no-load power

consumption of the feed gear with and without traveling support, and the power consumption of the support alone.

7 In the tests the total power consumption of the feed gear was taken as motor input times motor efficiency, the latter having been accurately determined for all conditions of load and speed.

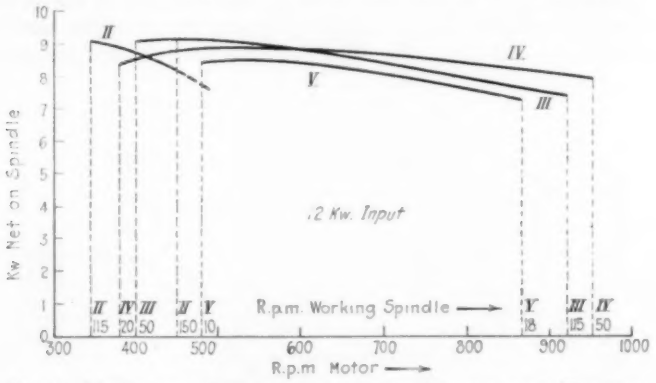


FIG. 3 MECHANICAL POWER DEVELOPED ON SPINDLE WITH VARIOUS GEAR RATIOS

8 *Losses in Headstock.* Careful tests by means of a prony brake with a cast-iron disk mounted on the faceplate of the lathe gave exact data with regard to the relation of actual power developed on the faceplate and electrical input. Fig. 3 shows the actual

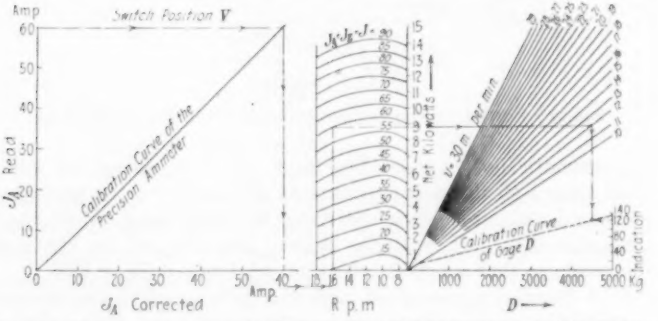


FIG. 4 CURVES FOR MUTUAL CHECK ON MECHANICAL AND ELECTRICAL READINGS

power developed with various gear ratios and at various motor speeds, but with a constant electrical input of 12 kw. The greater losses with gear ratio V are due to higher gear pressures; the drooping characteristics in III and V, and particularly II, are caused by friction losses due to high speeds.

9 In order to have a ready check on the accuracy of all readings, a set of curves was designed as reproduced in Fig. 4. These curves show immediately the relation which should exist between readings of the ammeter and the indications of the gage showing the *D* (or vertical) component.

10 At the left the readings of the ammeter are corrected according to its calibration curve. In the center are curves which give for any input current and spindle speed the corresponding power output at the faceplate on the basis of 220 volts. The fan-like set of

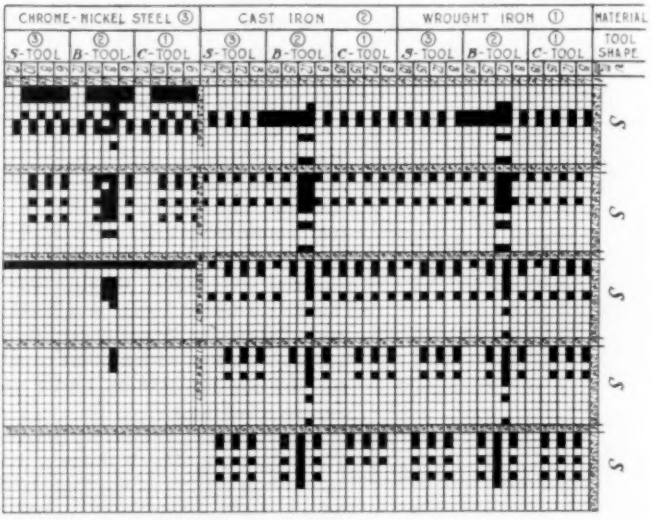


FIG. 5 PROGRAM OF CUTTING TESTS

curves in the right section gives the vertical component of the lip surface pressure corresponding to the power on the faceplate and to various cutting speeds. By means of the calibration curve below the curves it is possible to find the corresponding indication of the vertical pressure gage.

11 The use of these curves is simple, and with their help any error that might occur can immediately be detected and the cause determined before time is wasted by further tests on the basis of defective equipment or erroneous assumptions.

CUTTING TESTS

12 *Program of Tests.* The tabulation in Fig. 5 shows the tests made on different materials, namely, wrought iron, cast iron, and chrome-nickel steel, with three different forms of tools shown in Fig. 6, using different cutting speeds and many different chip sections. This tabulation indicates clearly the large variety of

tests made and the wide range of conditions over which they extended.

13 It was aimed to determine the influence, upon the cutting forces, of

1 Cutting speed
2 Chip section
3 Shape of tool
4 Characteristics of materials (wrought iron, cast iron, and chrome-nickel steel).

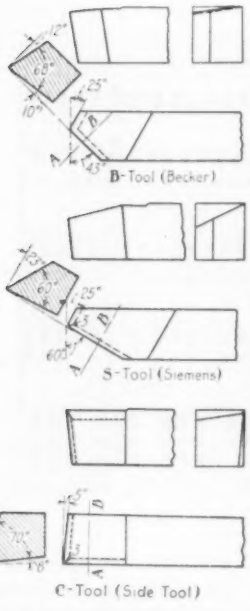


FIG. 6 SHAPE OF TOOLS USED

14 *Tools Used.* Fig. 6 shows the three tools used in these tests. The tool designated as tool *B* (or Becker) is one which was used by Schlesinger and Kurrein in their tests in 1913 to determine the life of tools. The angles here used represent averages as customary in machine shops abroad. The angle of 68 deg. is the one considered most favorable by Taylor. Tool *S* (or Siemens) is one favored by some manufacturers. Tool *C* (or side tool), is a form of tool having a large angle at the point, i.e., a tool with a point capable of resisting destruction under very severe conditions.

15 *Influence of Cutting Speed on Chip Pressure.* Fig. 7 shows a number of test results at various cutting speeds for wrought iron, cast iron, and chrome-nickel steel. Generally speaking, there is a tendency for chip pressures to decrease slightly with increasing cutting speeds. This conclusion agrees with that reached by Taylor and by Nicolson.

16 Tests, not shown in these curves, at 197 ft. per min. show only a slight decrease in pressure as compared with tests under similar conditions at 59 ft. per min.

17 *Influence of Chip Section on Chip Pressure.* Before proceeding with a discussion of the influence of the chip section and of the form of tool, attention should be called to the difference between the nominal section (i.e., depth of cut × feed) and the actual section removed. The relation between the two depends on feed, depth of cut, form of tool, and angle under which the tool is applied.

18 Fig. 8 shows a longitudinal section through a shaft after it has been turned off, on the basis of a certain shape of tool and

feed. Assuming that the shaft had previously been turned with a small enough feed so that its surface was practically smooth, it will at once be apparent that the chip removed is smaller than its nominal cross-section $t \times s$. To assist toward an analysis of the situation, Fig. 9 shows two successive positions of the same tool at an interval of one complete turn of the shaft, assuming various positions of the tool and a smaller or larger feed. The line showing the leading cutting edge in one position and the line

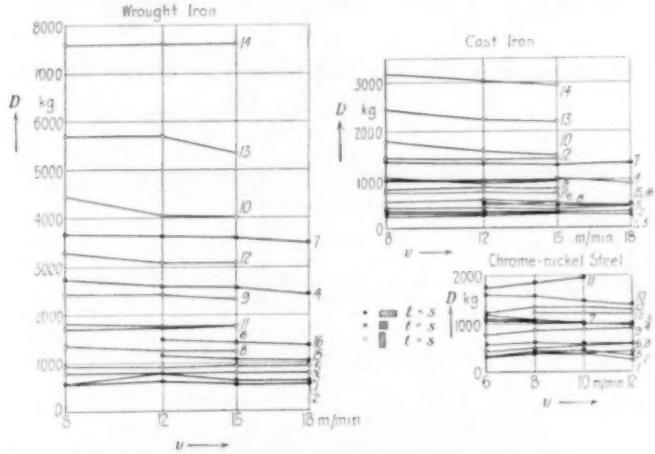


FIG. 7 CHIP PRESSURES AT VARIOUS CUTTING SPEEDS

showing the rear cutting edge in the next position intersect at point A. It is at once apparent that an area remains unaffected by the tool which depends on the feed s, the distance h of the point A from a line connecting the deepest points of the grooves, by the radius r at the point of the tool and by the two angles α and β. This unaffected area must be deducted from the nominal chip section $t \times s$ in order to obtain the actual area removed.

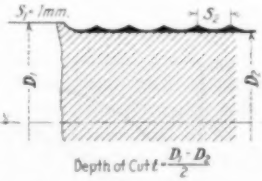


FIG. 8 SECTION OF SHAFT AFTER TURNING

19 The actual area removed may be expressed in percentage of the nominal chip section and a curve may be drawn giving this percentage for various nominal chip sections on the basis of a certain tool, a certain angle of application, and a certain feed. Fig. 10 shows a set of such curves for each of the three tools used. The assumption is made that the shaft had previously been turned with a 1-mm. feed. The curves indicate plainly that with the larger feeds the percentage of actual section removed be-

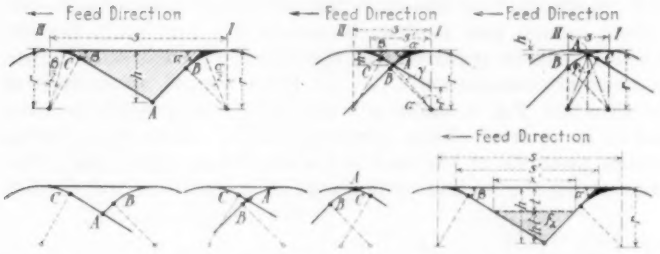


FIG. 9 INFLUENCE OF FEED AND TOOL POSITION ON SECTION REMOVED

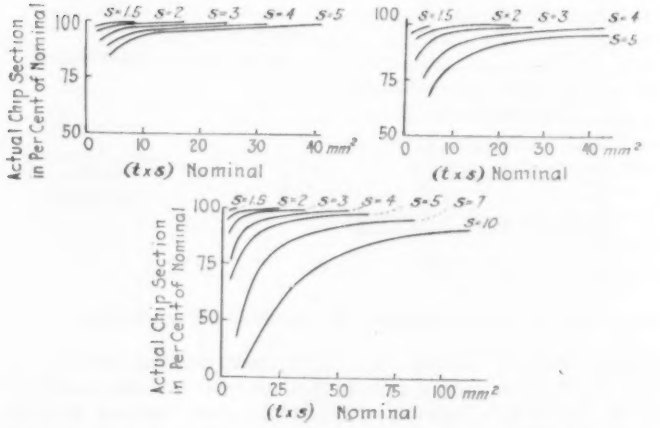


FIG. 10 RELATION BETWEEN ACTUAL AND NOMINAL CHIP SECTION

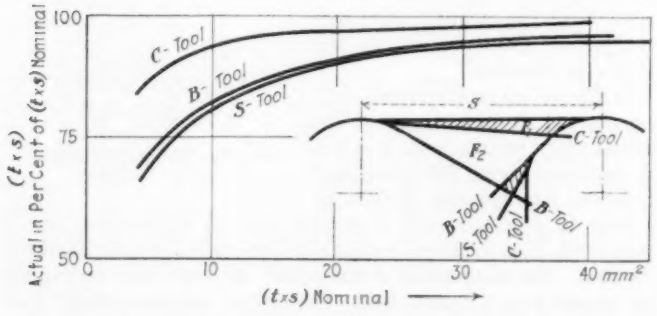


FIG. 11 COMPARISON OF THREE CURVES OF FIG. 10 FOR A 5-MM. FEED

comes smaller, other things being equal. The curves also show that with greater depths of cut the sections removed become larger in relation to the nominal section, other things being equal. With a 1-mm. feed the curve would, of course, become a horizontal line

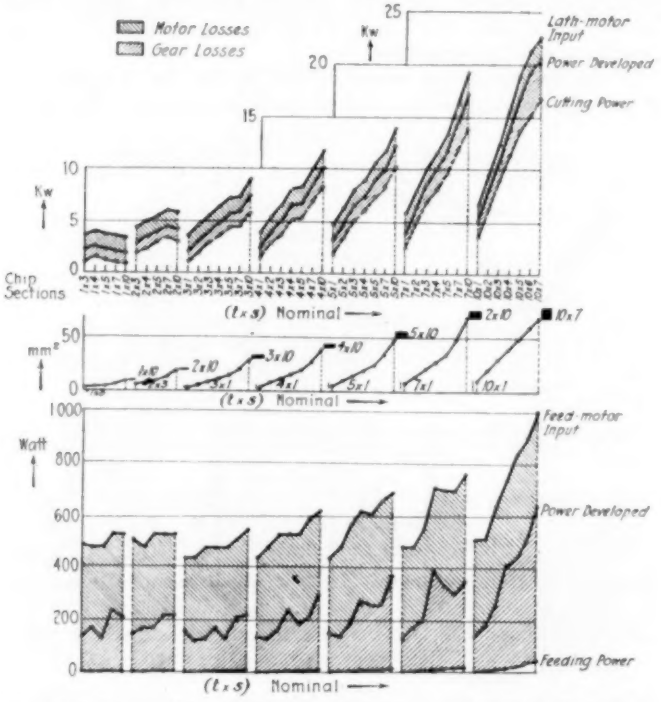


FIG. 12 POWER REQUIREMENTS AND LOSSES FOR 45 CHIP SECTIONS

located at 100 per cent. Fig. 11 shows a comparison of the three curves for the feed $s = 5$ mm. of Fig. 10.

20 Some of the results obtained in the many tests made are given in Figs. 12 and 13, which show a few characteristic sets of curves for 45 different chip sections. In the upper part of Fig. 12, the nominal chip sections are found on the abscissa according to depth of cut and feed, while input, losses, and actual power requirements are given in each case by the ordinate. Below these curves are shown the corresponding values for the feed gear and its electric motor. The low efficiency of this feed gear is striking, although in this particular case it is made more so by using a separate electric motor which is larger than necessary.

21 In the left-hand part of Fig. 13 will be found the power requirements for chips of certain corresponding cross-sections, all

comparisons being based on nominal cross-sections. It will be seen that a chip of 1 mm. depth of cut and 10 mm. feed will require only about one-half the actual power as compared with a chip of 10 mm. cut and 1 mm. feed. Fig. 13 also shows the distribution of losses in the lathe for four particular chip sections, namely, 5 x 10 mm., 10 x 5 mm., 2 x 10 mm., and 10 x 2 mm., these dimensions also being nominal.

22 In the succeeding curves, shown in Fig. 14 and following, all pressures relate to actual chip sections, not to nominal ones, and

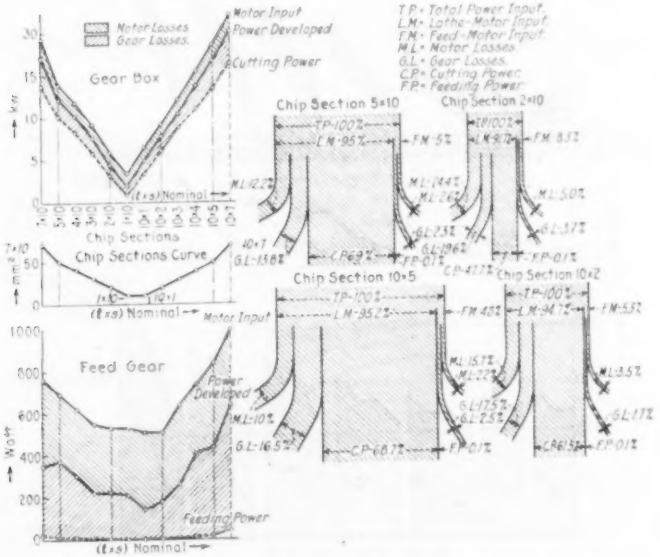


FIG. 13 POWER REQUIREMENTS AND LOSSES FOR TWO SETS OF CORRESPONDING CHIP SECTIONS

the losses in motor and lathe are not shown. However, all results taken with the dynamometer support have been checked on the basis of motor input and losses in motor and lathe, as previously established. In the curves the vertical pressure components are given for the different forms of tools and for the three different materials, and the effect of increasing chip sections on this component, and therefore on the power requirements, will be clearly visible. It will be seen that with increasing chip sections the power requirements do not follow the same law with different materials. In all cases, however, the curves show that power requirements do not increase proportionally to chip section but more slowly. Attention should here be called to the fact that in Fig. 18, discussed later, the shallow chips show consistently lower power requirements than do the deep chips. This observation,

already referred to, agrees with the results of other investigators, particularly Taylor, who made tests with cuts up to 1 in. deep.

23 *Influence of Shape of Tool on Chip Pressure.* In Fig. 14 are given three curves based on results with tool *B* and covering three materials. Individual test results are shown for the other

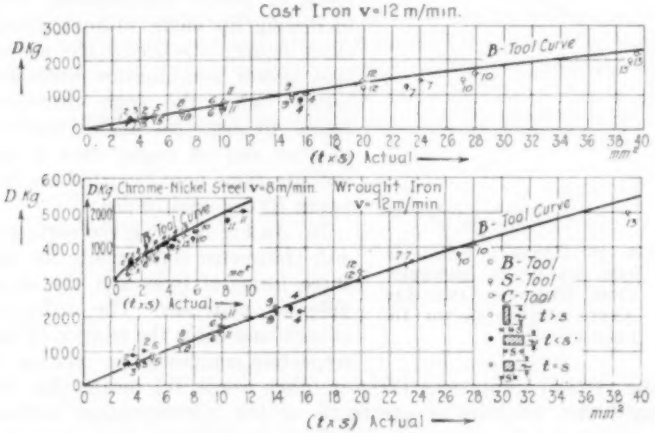


FIG. 14 CHIP PRESSURES WITH THREE SHAPES OF TOOLS CUTTING THREE KINDS OF MATERIAL

(Results with *B* tool in solid lines; others given individually.)

two shapes of tools. Cutting speeds were about 40 ft. per min. with cast and wrought iron and about 26 ft. per min. with chrome-nickel steel. The results obtained do not prove a material difference in the behavior of the three tools used.

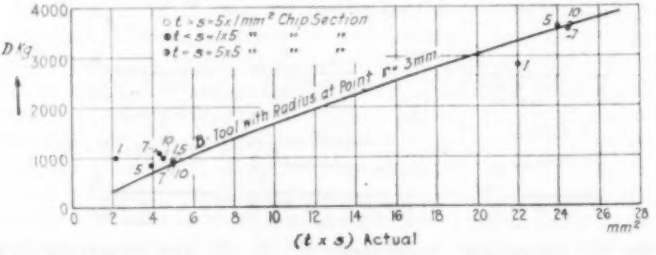


FIG. 15 EFFECT OF RADIUS AT POINT OF TOOL UPON CHIP PRESSURE

(Solid line for 3-mm. radius. The figures indicate the other radii in mm.)

24 Fig. 15 shows the effect of changing the radius at the point of tool *B*. The solid curve is for a radius of 3 mm., while the figures at each point indicate the other radii used in millimeters. The two results with 1 mm. should be disregarded as unreliable since the point did not stand up long enough to allow a reliable

reading being taken. It will be seen that the radius at the point has no appreciable effect. The results secured with an average radius such as used in these tests, may therefore be considered as applying generally.

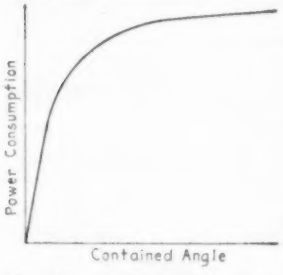


FIG. 16 GENERAL TREND OF THE RELATION BETWEEN CHIP PRESSURE AND LIP ANGLE ACCORDING TO DE LEEUW

25 Fig. 16 indicates the general trend of power consumption with increasing lip angle without giving exact figures. DeLeeuw has shown that power consumption decreases rapidly when the angle decreases below 40 deg. All angles used in practice and all angles used in the investigations here reported are between 50 deg. and 70 deg.

26 In Fig. 17 are given surfacing and transverse components for the three forms of tools and the three different materials. The ordinates in each case show the relation of the respective component in relation to the vertical component (D); in other words, 0.1 means that the particular pressure is one-tenth of the corresponding vertical component. In all tests the B-curve (transverse component)

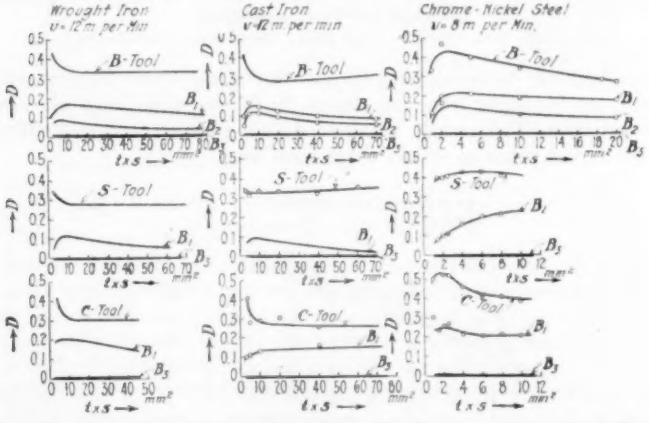


FIG. 17 TRANSVERSE COMPONENTS B_1, B_2, B_3, AND SURFACING COMPONENTS (C) FOR THREE TOOL SHAPES (B, S, C) AND THREE KINDS OF MATERIAL CUT

(The ordinates represent decimal parts of the corresponding vertical component D. B_1, B_2, B_3 respectively correspond to chip sections with t/s greater, equal to, or less than unity.)

was highest for deep chips and lowest for shallow chips. With the latter ones, in fact, the B-curve is sometimes located below the axis of abscissas, indicating that the remaining material on the

shaft exerts a pressure in the direction of the feed. The B_1-curves refer to chip sections with t/s larger than 1, the B_2-curves to those with t/s equal to 1, and the B_3-curves to those with t/s smaller than 1.

27 *Influence of Characteristics of Materials on Chip Pressure.* Reference should be made to Figs. 14 and 17 which give results with the three different materials. Fig. 18 shows vertical pressures (D) and specific vertical chip pressures (D_{sp}) for the three materials. The curves represent average results, but reference has previously been made to the location of points above and below these averages depending on the particular shape of the section.

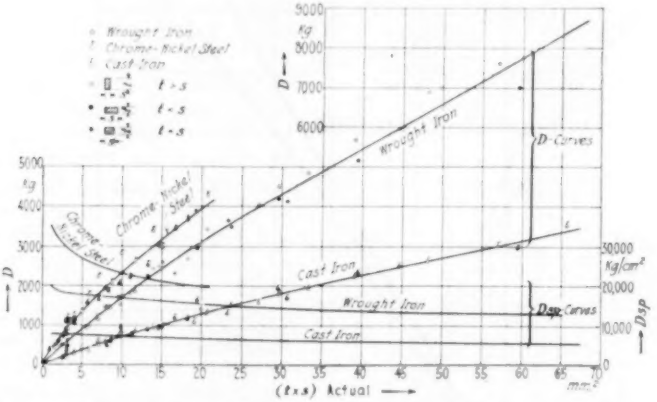


FIG. 18 VERTICAL PRESSURES (D) AND SPECIFIC VERTICAL PRESSURES (D_{sp}) WITH THREE MATERIALS

28 Figs. 19 to 21 give, on the basis of nominal chip sections, comparative results obtained by several other investigators and the author, each figure relating to one kind of material. There is also given a curve properly corrected to cover actual chip sections. Fair agreement exists among all the test results except with some of those obtained by Nicolson.

29 In an attempt to reduce chip pressures to a formula, the author followed the method adopted by Meyer in 1908 in drawing Brinell-hardness curves on double logarithmic paper, using as the abscissa the area of the Brinell impression and as the ordinate the pressure applied. The author drew similar curves with the vertical component of the chip pressure as ordinate and the actual cross-section of the chip as abscissa. The curves under the circumstances become straight lines, the same as the curves drawn by Meyer. This at once suggested a comparison of the two sets of curves for the three materials used in these tests. When the hardness curves were added, it was found that they formed a

larger angle with the axis of abscissas, and that they intersected with the chip-pressure curves. (See Fig. 22.)

30 There is an interesting difference in the relative location of the Brinell hardness curves and of the three chip-pressure curves. The Brinell curve for cast iron is between those for chrome-nickel steel and wrought iron, while in the case of chip pressure, the curve for cast iron is considerably below both of the other two curves. This shows that cast iron, notwithstanding its great hardness, is comparatively easy to work on a lathe or planer.

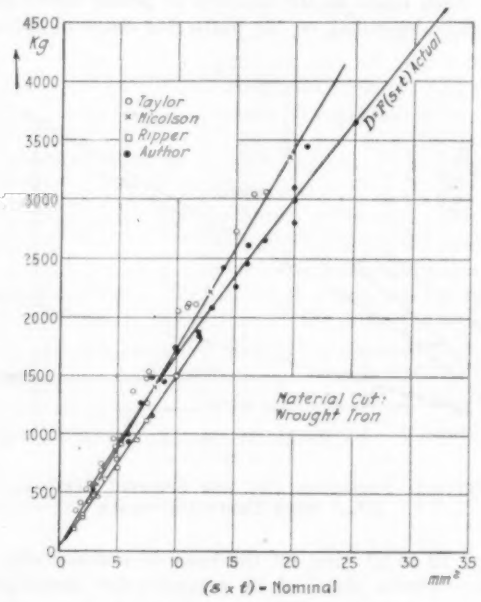


FIG. 19 COMPARISON OF CHIP PRESSURES AS MEASURED BY VARIOUS INVESTIGATORS

31 Chip-pressure curves and the corresponding Brinell curves are drawn for six different materials in Fig. 23. They are supplemented by a set of curves for high-carbon steel consisting of the Brinell curve and pressure curves in accordance with the results obtained by Kurrein in drilling tests. It appears that the intersecting points of the hardness curves and chip-pressure curves will be found near two approximately parallel straight lines, one located considerably below the other. The upper line connects the intersecting point of curves relating to materials forming continuous chips, such as steel, wrought iron, copper, etc., while the lower line connects the intersecting points of curves relating to materials such as cast iron and brass.

32 These observations permit the determination of chip pressures and therefore power requirements in the turning and planing of materials of which the Brinell-hardness characteristics are known.

The Process of Metal Cutting

33 *The Formation of a Chip.* It is surprising that the actual formation of a chip has been studied but little, with the result that rather vague ideas still prevail regarding this fundamental operation. A clear observation of a turning tool working on a lathe is practically impossible, since, during the process of turning, the point is imbedded in the material to such an extent that the exact action of the tool cannot be seen. The author therefore decided to use a shaping machine and a work piece which was planed and finished on one side so that all fissures and all compression of material could be well observed. The tool was set so that it worked on this finished surface. Since, however, in a shaping machine the tool moves, and since it appeared desirable to photograph individual phases of the process, a planer was later substituted which was operated by hand so that it could easily be stopped at any point desired.

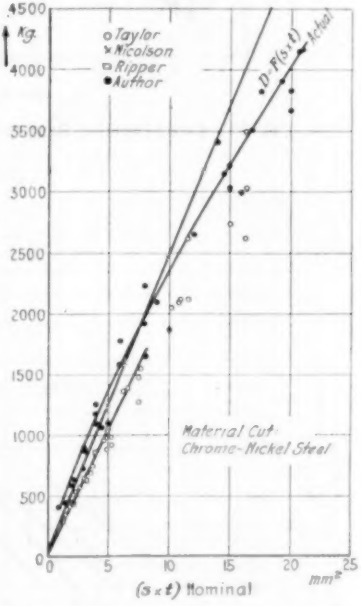


FIG. 20 COMPARISON OF CHIP PRESSURES AS MEASURED BY VARIOUS INVESTIGATORS

34 The two sets of pictures shown in Fig. 24 refer to two different sizes of chips, but they both show about the same successive steps. Referring first to the left row, it will be noticed that in the first picture the point of the tool compresses a small quantity of material. In the second picture this compression has gone so far that a large tear has been caused between the main part of the work piece and the chip element being separated from it. This tear extends from the point of the tool to a point on the surface of the work piece a small distance in front of the tool. In the succeeding pictures the process described repeats itself. In the fourth and fifth pictures of the left-hand row small fissures are noticeable under the point of the

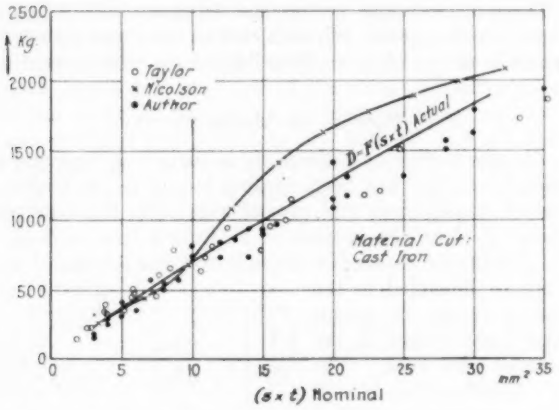


FIG. 21 COMPARATIVE RESULTS OF CHIP-PRESSURE TESTS

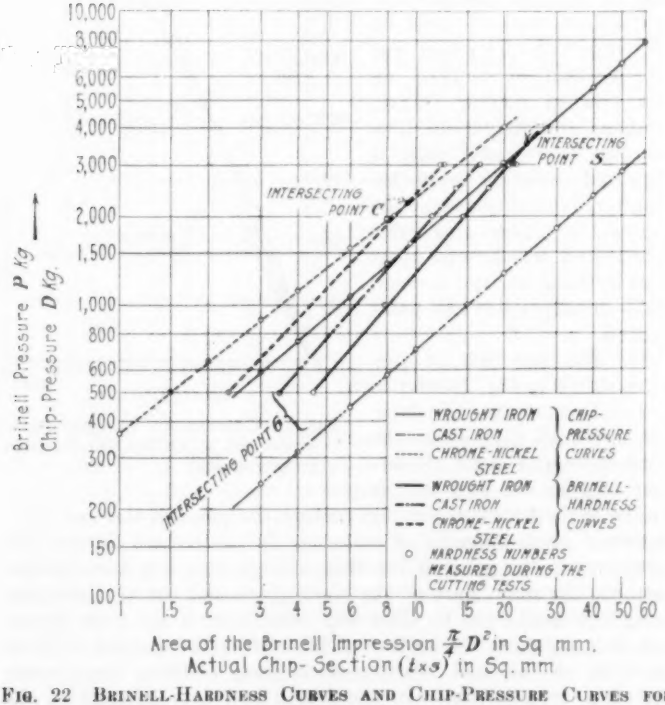


FIG. 22 BRINELL-HARDNESS CURVES AND CHIP-PRESSURE CURVES FOR THREE MATERIALS

tool, a phenomenon to which, as far as the author knows, no attention has been called heretofore.

35 The second row of pictures shows similar steps, but more clearly since a larger chip was being taken and since the photographed steps follow each other more closely.

36 In Fig. 25 the formation of a continuous chip is shown. Such a chip does not show subdivision along the side of the chip adjacent to the lip surface, but the individual chip elements may be clearly recognized.

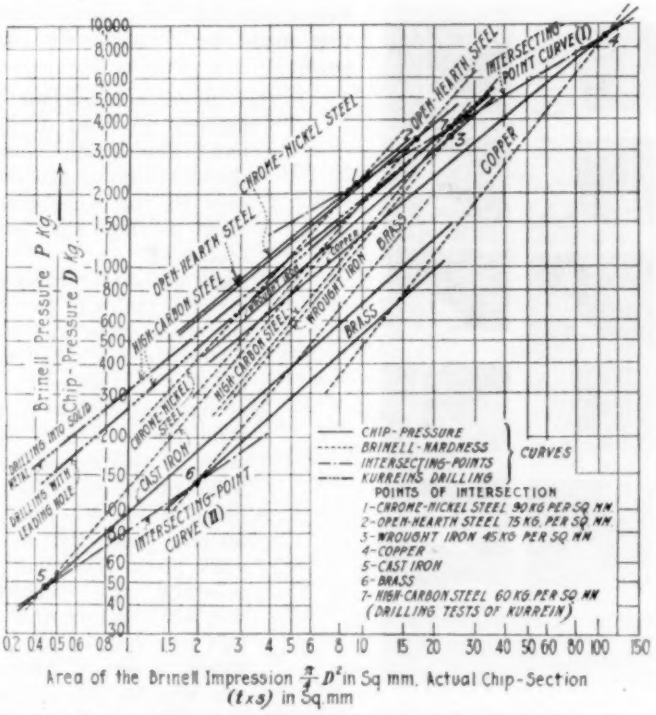


FIG. 23 BRINELL-HARDNESS CURVES AND CHIP-PRESSURE CURVES FOR SEVEN MATERIALS, INCLUDING KURREIN'S DRILLING TESTS

37 In order to show that chips in turning and planing have the same characteristics, both kinds were photographed in plan view and side view (see Fig. 26). The similarity of both chips leaves no doubt that the process in turning is the same as that in planing, notwithstanding the fact that the two chips were taken at entirely different cutting speeds. The surfaces from which the chips were cut also show similar appearance. The lower picture in Fig. 27 shows the surface from which the one set of chips was turned at a cutting speed of about 27 ft. per min., while the one above shows

the surface obtained in the test on a planer. Both pictures show very similar lines across the surface. The cutting edge of the planer tool used was inclined at an angle of 5 deg. to a line vertical to the direction of motion, and the fissures in the corresponding surface show a similar direction. The individual fissures indicate the points at which the cutting edge started a new operation after a chip element has been broken off, or probably a point slightly later than this starting point.

FIG. 24 SUCCESSIVE STEPS IN THE FORMATION OF CHIP ELEMENTS

38 From what has been said, it would seem that the process of cutting might be divided into four parts:

1 The initial compression of material in front of the tool. This action causes the tool to get its first new grip on the work piece after a chip has been broken off

2 A slight initial tear is caused between the main part of the work piece and the material which is to form the next chip element

3 The material which is to form the next chip element is compressed and bulges on the lip surface

4 A tear is caused between this material and the main work piece. This tear is generally not complete but, with the exception of cast iron and similar materials, a certain connection remains between the individual chip elements. This connection makes a chip of the individual elements broken off.

39 The conception here outlined explains the oscillating form of chip-pressure curves obtained by Nicolson and others. In these curves the points of maximum pressure represent the moment of maximum compression of the chip elements, while the lowest points immediately succeeding the points of maximum pressure represent the moments after the chip elements have been broken off.

40 A very similar conception to that here outlined is contained in a sketch by Codron, drawn by him without having photographed the successive stages and reproduced in Fig. 28. The agreement between Codron's conception and the one here given is of par-

ticular value for the reason that at the time the present tests were made, Codron's sketches were unknown to the author. The pressure curve obtained by Codron and seen at the right of Fig. 28 agrees closely with that obtained by Nicolson.

41 The subdivision into chip elements is not always as noticeable as in the pictures previously given. Fig. 29 shows a chip consisting partly of clearly separated chip elements, and partly of a rather continuous section in which the individual chip elements can hardly be recognized.

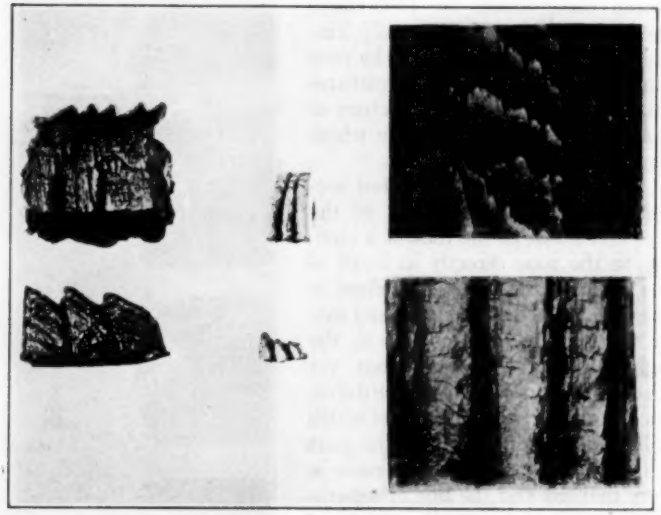
FIG. 25 FORMATION OF A CONTINUOUS CHIP

42 *Structural Changes in the Chip.* Professor Kurrein's microscopic observations in 1905 have been confirmed and materially amplified by the author. From the very large number of microscopic pictures taken to investigate the structure of a chip, a few are given here which appear of particular interest.

43 Fig. 30 shows a vertical section parallel to the motion of the tool and taken at the root of a chip, i.e., in the zone directly in front of the point of the tool. The effect of the tool, located at the left-hand side of the picture and moving to the right, upon the material not yet severed, is clearly visible. The difference in appearance of the part which will next be detached and the part which remains on the work piece is very marked and the line of separation at once apparent. Fig. 31 shows a point of the chip element in contact with the lip surface of the tool above and adjacent to Fig. 30.

44 *Effect of Chip on the Tool.* Aside from the effect of the turning process upon the cutting edge proper consisting mainly in destruction by impurities contained in the material, there is a gradual effect of the chip upon the tool which in time affects the cutting edge. This latter action is radically different from the former; it does not start at the edge but on the lip surface, a slight distance back of the edge, and causes a cavity which in time becomes larger, extending more and more toward the rear but also slowly approaching the cutting edge. When it comes in contact with the latter, it breaks it down completely at the point reached. This process will become entirely clear by reference to Fig. 32 in which 1, 2, and 3 show the successive stages of destruction.

45 The phenomenon mentioned is most noticeable with heavy chips but even with small chip sections it can be observed, especially when turning material of great toughness. Fig. 33 shows the lip surface of a tool used for chrome-nickel steel with chips of only 1 mm. depth and 3 mm. feed. The result is very marked. The cavity begins about a sixteenth of an inch back of the cutting edge, but the edge itself is not yet injured. This important action of the chip has been given surprisingly small attention heretofore. It is clearly a result of the great compression of the chip element in formation and of the friction between the chip



FIGS. 26 AND 27 COMPARISON OF CHIPS ON PLANER AND ON LATHE
(The surfaces from which the chips were taken are shown at the right.)

and the lip surface upon which it slides. With larger chips, as mentioned before, the effect becomes even more marked, and incidentally the distance from the cutting edge at which the cavity starts is slightly increased. With wrought iron this effect is less marked; with cast iron it is not noticeable.

46 As a result of these observations a new form of tool has been designed which is intended to reduce to a minimum the amount of compression and deformation of the chip and the marked cup action on the lip surface coincident with it. At the Spring Meeting of the Society at Cincinnati in 1917, Mr. DeLeeuw reported the results of certain investigations with a tool which, in fact, was an exact copy of a worn tool, in other words, a tool which embodied a depression in the lip surface exactly like that

produced by the chip. It is evident that if a tool can be designed which will keep down to a minimum the expensive and detrimental work of the chip on the lip surface, which will reduce materially the compression of the chip, which will have a long life, and which, at the same time, can be produced commercially, a great deal will be gained. Fig. 34 shows one form of tool provided with the cutting edge developed by the author. The tool here shown has a front and rear cutting edge and has along both of these cutting edges a narrow raised strip projecting above the rest of the lip surface. The connection between the depressed section and the raised strip is made by means of a fillet, with the result that the cutting edge has a strong support and that the heat is efficiently conducted away from the cutting edges. The lip angle is generally chosen in the neighborhood of 70 deg., which also favors the conduction of the heat away from the cutting edge proper.

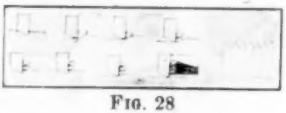


FIG. 28

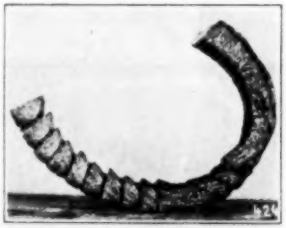


FIG. 29

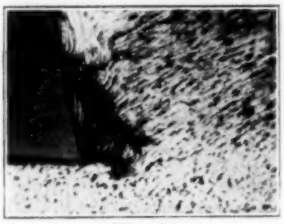


FIG. 30

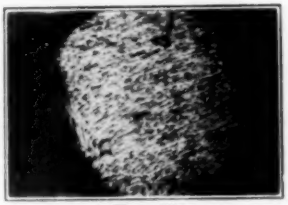


FIG. 31

47 From what has been said before, the reduction in the chip pressure upon the lip surface, which has been definitely proven, will not appear surprising. It is furthermore believed that a favorable action of the raised strip upon the chip element in formation takes place due to the fact that the rear edges of the raised strips exert a pressure upon the chip element in formation that is favorable to its separation. This effect appears likely from the fact that the rear edge of the raised strip is gradually attacked in operation and wears away just as previously shown in connection with the cutting action in a continuous lip surface. (See also Fig. 35.)

48 The characteristics outlined permit of higher cutting speeds or larger chip sections, or both, than is practicable with the standard tools now used. As a result of this, production is increased and this is done without straining the machine tool used, since

even with heavier chips the chip pressure is not increased and since with the same size of chip the chip pressure is reduced so that even a higher cutting speed will not mean greater power consumption.

FIG. 32 EFFECT OF CHIP ON LIP SURFACE

FIG. 33 LIP SURFACE GUTTERED OUT BY THE CHIP

FIG. 34 KLOPSTOCK CUTTING EDGE

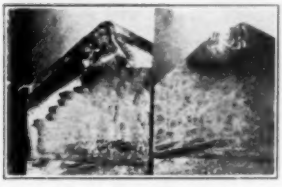


FIG. 35 WEAR OF THE KLOPSTOCK AND OF THE STANDARD TOOL

49 Extensive tests have been made with many different and even abnormal chip sections in order to test the working and the life of this cutting edge. Results obtained vary with conditions as was to be expected, but as an average it may be stated that with chrome-nickel steel the vertical component is reduced about 26 per cent and with wrought iron, 14 per cent. The transverse force is reduced at an average about 65 per cent, the surfacing force 60 per cent with chrome-nickel steel and 50 per cent with wrought iron. The difference between the vertical component and the total chip pressure is only about 2 per cent.

50 A noticeable difference exists with regard to the exact saving in power consumption with different chip sections. With chips causing a considerable amount of material to be taken off by the rear cutting edge, the material flowing over this rear cutting edge meets the material coming from the front cutting edge and a certain amount of interference and deformation takes place as a result thereof. These are the greater the smaller the angle between front and rear cutting edges, and an angle less than 90 deg. is therefore inadvisable. On the other hand, a chip section which causes material to be cut almost entirely by the front cutting edge will allow the chip to flow freely without interference. The chip sections of great depths and small feed therefore show a greater saving than those of small depths and large feed.

51 The difference in the manner of wear in operation is clearly shown in Fig. 35. It represents two tools made of the same tool steel and operated under similar conditions, in this case abnormal

conditions. The cutting speed was about 197 ft. per min. The standard tool shown in the picture to the right shows the edge near the point destroyed while the tool with the raised strip shows destruction of a short section of the narrow strip which is clearly seen to have started in the rear. The point in this latter case is still in fair condition.

52 Figs. 36 to 38 show the different appearance of chips produced by standard tools and the new tool. The right-hand column of Fig. 36 shows chips produced with a standard tool. The spiral at the top represents the chip cut with a newly ground tool; the picture below shows the chip with the tool beginning to get dull, and the small piece at the bottom represents the sections torn off with the dull tool. The time during which this last kind of chip was produced in these particular tests represented more than half that during which the tool was used, and practically during the whole period when short chips were produced a chattering noise was heard.

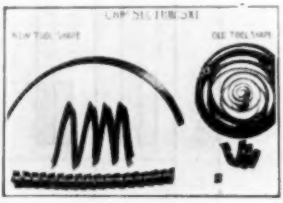


FIG. 36

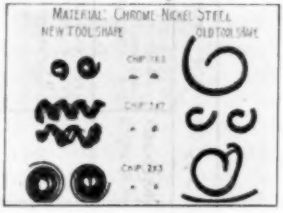


FIG. 37

53 The left-hand column shows chips produced with the new tool. At the beginning chips are produced of a very large radius. This radius gradually decreases as the cutting edge is affected and still later a long helix of small radius appears. In most cases this shape of chip continues until finally the fact that the tool has become dull is noticed by the chattering noise. The two other figures mentioned show chips of various cross-sections and various materials. It will be noticed that the new tool produces a much more continuous chip, subdivided into individual chip elements to a much smaller degree. The most striking difference between the two kinds of chips will be seen from the comparative cross-sections of the chips which are found in the center of these pictures. With the same depth of cut and with the same feed the chip produced by the new tool has a very much smaller cross-section than that produced with the standard tool, which is of course due to reduced compression and deformation in

FIG. 38

FIGS. 36, 37 AND 38 CHIPS WITH KLOPSTOCK AND WITH STANDARD TOOLS

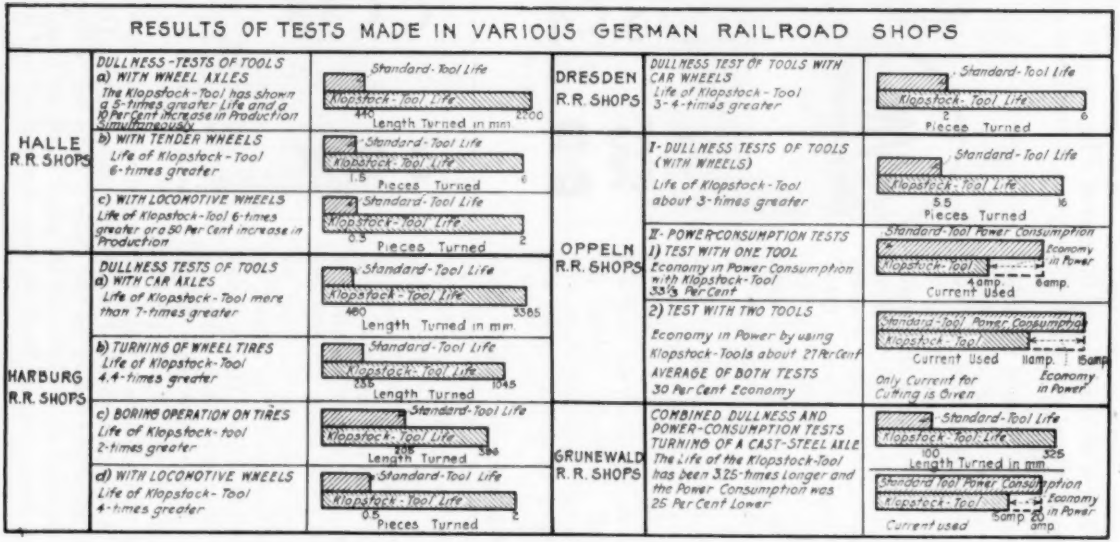


FIG. 39 RELATIVE LIFE OF KLOPSTOCK AND STANDARD CUTTING EDGES

the case of the chip produced by the new tool. It goes without saying that in the latter case the chip runs off at a correspondingly higher speed.

54 An extremely interesting fact in connection with this is that the chips produced by the standard tools are of dark color while those produced by the new tools under exactly the same conditions are very light in color. The difference in temperature has under certain conditions been found to be as much as 100 deg. cent. or 180 deg. fahr.

55 The material reduction in the heat produced, and the efficient manner in which this heat is conducted away from the cutting edge, leave the latter in a good condition for work up to the last, with the result that the surface turned by means of the new tool is generally better than that turned with the standard tool.

56 A most important point with any tool is, of course, its life. A large number of tests were made to determine the relative life of the new and standard types of tools. It was invariably found that under the same conditions the new tool lasted very much longer than the standard tool, which no doubt is a result of the combination of a large lip angle with a reasonably low temperature in operation. In other words, the new tool has a cutting edge which is mechanically strong on account of a large lip angle, but which remains reasonably cool, first, because the heat produced is less, and, secondly, because the design of the tool does not interfere with efficient conduction of the heat away from the cutting edge.

57 Fig. 39 presents the results of some of the tests made in various German railroad shops. Hundreds of similar tests have been made which have given similar results. The conditions prevailing in the tests here tabulated were exactly the same in the case of both tools, and only the relative life of the cutting edge was compared.

58 The manner of producing the new form of tool has been given careful study. Complete tools or cutting tips for welding to shanks are generally made in dies by hammering or pressing. If made in this manner, the tool need only be ground along the narrow raised strip and on the clearance surface. The depressed part of the lip surface is in this case left untouched. No difficulties are encountered in the tool room; in fact, it is possible with the new type of tool to reduce materially the number of forms of tools. Foreign shops have found it entirely practicable to get along with a very few standard forms, owing to the fact that cutting conditions are more favorable with the new tool.

59 In normal operation the cutting edge can be reground by the operator four times, after which it must be newly forged. The tool is of course hardened in the same heat in which it is

forged, i.e., it is first forged and then its temperature is brought exactly to the proper hardening temperature.

60 If it is desired to produce the depression in the lip surface entirely by grinding instead of forging, this can readily be done by means of a special attachment to a grinder equipped with a small-diameter grinding wheel as shown in Fig. 40. The attachment permits of easy adjustment of the tool to the proper angles, and the regrinding of a depression which has become too shallow can, by means of this attachment, be accomplished within a few minutes.

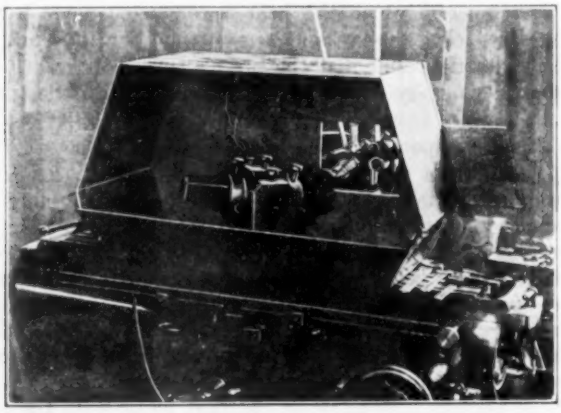


FIG. 40 GRINDING ATTACHMENT FOR GRINDING DEPRESSION IN KLOPSTOCK TOOL.

61 In estimating the effect of reduced chip pressures upon the amount of work which can be turned out on a particular lathe, the following should be kept in mind:

1 A decrease in chip pressure of, say, 20 per cent will permit an increase in the chip pressure of 25 per cent before the same chip pressure is again reached

2 A permissible increase in the chip pressure of 25 per cent means an increase in the area of the chip amounting to considerably more, the exact amount depending on the character of the material turned. On the other hand, an increase in the chip area of 45 or 50 per cent does not mean an increase in the production by a corresponding amount, since the time for setting up and removing the work piece must be taken into account

62 Generally speaking, results obtained in foreign shops have justified the belief that production can be increased about 30 per cent by means of the new tool.

DISCUSSION

A. L. DeLeeuw.[1] Without in any way desiring to deprecate the value of the experiments of the author of the paper, I wish to call attention to the fact that neither the essential observations nor the conclusions drawn therefrom are new, and I refer to a paper read by me before the Spring Meeting of the Society at Cincinnati, Ohio, in 1917. This paper bore the title A Foundation for Machine-Tool Design and Construction, and has been reproduced in Transactions for 1917.

On page 192 of Transactions, vol. 39, 1917, the following paragraphs will be found:

> 31 Forged spindles of sixty-point carbon steel were roughed by a tool as shown in Fig. 3 [Reproduced here as Fig. 41. — Editor]. As a rule, the tool was able to rough three spindles before a breakdown. In its broken-down condition the tool appeared as shown in Fig. 4 [Fig. 41]. A hollow had been ground out by the chip, but a land of a little more than $\frac{1}{64}$ in. in width had been left at the front end, showing that the extreme front of the tool had not been in action. The experiment consisted of carefully measuring the broken-down tools and making new tools of just that shape; in other words, a tool like the old tool, but with a hollow ground in the top of the same shape, size, and location as in the old tool.
>
> 32 This tool is shown in diagram in Fig. 4 [Fig. 41]. The hollow was carefully polished, and a tool thus prepared would rough from 9 to 13 spindles. Examination showed that the hollow in the tool would remain smooth almost to the last, and that a complete breakdown followed very soon after the surface of the hollow began to show scratches. No tests of power consumption were made, but it may be assumed that the power required with the old tool was more than with the new tool, as the chip did not have to bend so sharply and as the work required for hollowing out the tool was omitted.
>
> 33 Another interesting point about this tool was that the actual contained angle between the front of the tool and the front of the hollow was much less than we would have dared to make between the front and top of an ordinary lathe tool, especially if this lathe tool were to be used for roughing. Nevertheless, under the conditions given, this tool with the small front angle stood up better than the original tool with the large angle.

It will be noted that in all essentials the tool described there is the same as that described now by Dr. Klopstock.

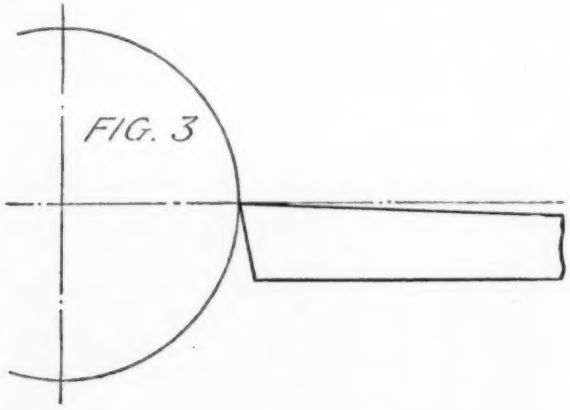
The experiments to which I refer were made in 1912 and 1913 and since then have been described on various occasions, both in print and in lectures given by me.

I wish to call particular attention to the anomalous condition prevailing in our machine-tool industry which permits such experiments as those described in my paper in 1917 to go unchallenged and undeveloped, and perhaps even unnoticed, and allows the credit and benefits to go to investigators in a foreign country. I wish it understood that I am not blaming Dr. Klopstock or other investigators for taking up such a line of experimentation and

[1] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.

investigation, but that I do blame the American machine-tool industry and American colleges and universities for so completely overlooking or ignoring a lead such as I have given in my paper.

If this country wants to maintain its much-heralded lead in



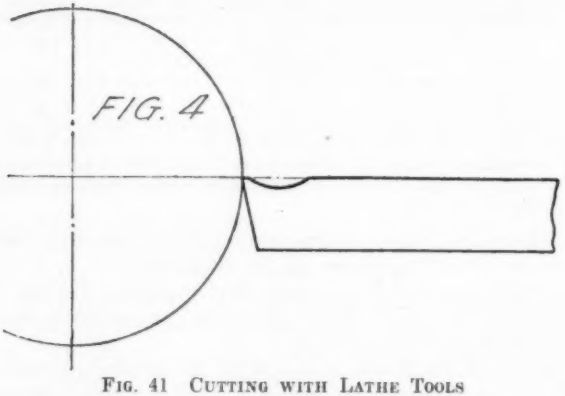


FIG. 41 CUTTING WITH LATHE TOOLS

machine-tool industry and machine-tool design, it is about time that somebody wakes up to the necessity of carrying out just the kind of investigations which Dr. Klopstock described in his paper and for which he deserves full credit, even though the original and essential experiments were made before he started his.

JAMES A. HALL.[1] The author of this paper is to be congratulated on the extent of his investigations and on the care and detail with which they have been carried out. For instance, the checking of the power required as figured from the dynamometer readings with that given by the wattmeter makes one feel that the results are much more reliable than if the pressure dynamometer alone had been used.

The care taken to figure the net volume of the chip by analysis of the shape of the tool instead of simply multiplying the feed by the depth of cut is very interesting, but the writer is not sure from reading the paper if certain other errors have been taken into account. For instance, the work does tend to spring away from the tool, and this may affect the results. Moreover, under certain conditions the crack preceding the point of the tool in cutting may run in toward the center of the work and actually cause more metal to be torn off than would be given by a study of the tool contour. It seems desirable, at least on shallow cuts, to check the amount theoretically removed by weighing the chip.

With reference to the section beginning with Par. 33, it is interesting to compare Dr. Klopstock's analysis with the data given in a paper entitled Report on the Flow and Rupture of Metals during Cutting by Messrs. Rosenhain and Sturney, which was presented before the Institution of Mechanical Engineers in January, 1925.

Messrs. Rosenhain and Sturney made a study of chip formation by suddenly stopping cuts in a lathe and then taking in each case photomicrographs of polished and etched cross-sections of the chip and the metal from which it was being separated. As a result of this study they have classified chips into three types. First is the "tear" type of chip, where the metal is compressed. A crack starts at the point of the tool and runs in as the chip is being torn off, and finally this chip breaks by shearing and a new cycle is started. This is the type which Dr. Klopstock seems to have in mind, and is the one which would be expected with deep cuts and small rake angles. The second type given in the above paper is the "shear" chip, which continuously shears at an angle of about 30 deg. with the relative direction of motion of work surface and tool. This is apt to occur with shallower cuts and in metals which are much weaker in shearing than in tension. It is an intermediate form between the tear type and the third and generally most desirable form which Messrs. Rosenhain and Sturney have called the "flow" type. In this the chip is formed without any apparent tear crack ahead of the tool point and practically no shear cracks across the chip, but the action is somewhat analogous to what occurs in extrusion. These chips form on clearly defined angles with the

[1] Assistant Professor of Mechanical Engineering, Brown University, Providence, R. I. Mem. A.S.M.E.

relative motion of the tool and are more likely to form where the top rake of the tool is increased up to this angle. The pictures of the chips from the Klopstock tool seem to be the "flow" type, and the increased rake angle just back of the cutting edge may be giving the same result as the plain increase in rake angle used by Messrs. Rosenhain and Sturney.

Of course the power required by tools is such a small item in the expense of shop operation that few manufacturers are interested in it unless it is perfectly clear that there is a definite relation between this power and cutting life of the tool. The writer is pleased to note that Dr. Klopstock did not forget this point, but proceeded to design a tool with particular reference to the conduction of heat from the cutting edge and then had this tool thoroughly tried out in tests for endurance.

B. H. Blood.[1] What impresses me most is that the experimental or laboratory results have been combined with those of actual shop tests, as given in Fig. 39. Laboratory work is always to be taken with a grain of salt until it has the confirmation of practical shop use.

An instance in my own experience confirms Mr. DeLeeuw's criticism of our ultraconservatism in adopting new ideas. When the Maag gear system was first brought here some ten years ago, much the same shape was used in grinding cutters as illustrated by Dr. Klopstock in his paper. The face of the cutter was first ground flat and to the correct shearing angle, then it was concaved parallel to the cutting edge with a small emery wheel a quarter of an inch in diameter to give the shape exactly as described here. We had many of those cutters in use, and machines were developed and built for grinding the concave, but the practice was largely abandoned in the shop, probably without full realization of its value. One reason was that the same cutters were used alternately on cast iron, bronze, machine steel, forged steel, alloy steel, or steel castings, and when many kinds of material were being cut there was no one shape of tool best for all. The results, if they were good, were not followed out. I have to accept my share of criticism for that lack of appreciation of something which we had in our hands.

Referring to Dr. Rosenhain's paper, we have a point brought out there that this type of cutting edge is of greatest value where a crack is preceding the point of cutting tool; that is, where a chip is formed by tearing off and, therefore, the real edge of the cutting tool is doing but very little work. This is an ideal condition for long life of the tool. Just how much value it would have, if any, on a shearing cut I cannot foresee. If there is any information available on the subject which might be added to the paper, it would be useful.

[1] 30 Sherman St., Hartford Conn. Mem. A.S.M.E.

It would seem to me that on a very thin paring cut, the concave ground back of the cutting edge, with a narrow land between pocket and actual cutting edge, would lose its effect entirely.

O. W. BOSTON.[1] It is stated in Par. 4 that all measurements checked within plus or minus one-half of one per cent. Does this accuracy hold for the data for cast iron in Fig. 5?

In connection with some experiments at the University of Michigan along this line, we have had trouble in getting consistent results in the cutting of cast iron. We have gone to the trouble of running a heat, pouring the metal into one crucible, and then pouring from it all the test bars, so as to have as nearly as possible uniform test specimens. Even so, a very wide divergence from the accuracy claimed here was found. Presumably this accuracy applies only to the dynamometer, and not to the cutting force.

In Par. 14 there is a list of the tools used. There is great similarity between these tools and those used by Colonel Crompton in England, and described in somewhat detailed form in his report to the Institution of Mechanical Engineers in 1922. Colonel Crompton reported that his tool which corresponds to this Tool C had the least endurance of the tools which were used.

Did the author use any so-called round-nose tools in the comparison test with the Klopstock tools?

In Par. 24, Fig. 15, it is shown that a greater radius on the end of the cutting edge requires the same force as a tool having a sharper edge, but that it has more endurance. The writer believes that research to date will show that the balance is about even on that point; some show that the round-nose tools require greater force, and others, less force.

In Par. 22 the statement is made that the power requirements do not increase proportionately to the chip section, but more slowly. In experiments at the University of Michigan this problem has been divided into two parts: first, to determine for various chip widths the total force required on the tool to remove a chip of constant depth (The force was found to vary directly as the chip width.); and second, to determine for various depths of cut a relation between the total force and the depth of cut when the width is constant (It was found that the total force does not increase in direct proportion to the depth of cut, but less, which verifies the belief pointed out many times, that a thick chip is cut more efficiently than a thin one.) The chips were rectangular in cross section as from a cut-off tool and not complicated with a round-nose tool form. It therefore seems clear that chip area or chip section is not a proper basis for tool performance, and that

[1] Professor of Shop Practice, and Director of Engineering Shops, University of Michigan, Ann Arbor, Mich. Mem. A.S.M.E.

the chip should be expressed as a function of its thickness and depth.

It seems reasonable, therefore, in Fig. 18 that all of the points for a chip for which t is greater than s, should be higher than those for which t is less than s. Curves quite similar to these have been determined, but the point is brought up because it divides the effect of the thickness and the feed which is here combined as the cross-sectional area of the chip. This also agrees with the author's statement at the bottom of page 354, that "the shallow chips (with heavy feeds) show consistently lower power requirements than do the deep chips (presumably of the same sectional area)."

In Figs. 19 and 20 the writer questions some of the points for the same reason, that is, no information is given as to the exact values of the feed and thickness of cut. Taylor's results in these figures may be based on one ratio between s and t, while Nicolson's may be another. In order to have these curves mean something definite, the relation between the thickness and the feed should be given, rather than simply the area.

THE AUTHOR. With regard to Professor Boston's question as to the meaning of the statement in Par. 4 to the effect that all measurements checked within plus or minus one-half of one per cent, I wish to say that my remark referred to agreement between electrical and mechanical results of individual tests. As stated in the paper, mechanical and electrical readings were taken coincidently, and in checking one against the other, curves such as are shown in Fig. 4 were used which, as outlined in the paper, were developed from elaborate investigations regarding the detailed losses in the lathe on which the tests were made. Agreement of results of different tests was not within such close limits, but no greater variation was found with cast iron than with other materials. This may be seen, for example, in Fig. 18 where individual test results are indicated and the curves drawn as averages, and where individual results do not appear to vary from the average curve any more with cast iron than with other materials.

Professor Boston asks whether any tests were made to compare round-nose tools with Klopstock tools. A number of such tests were made and the results were similar to those obtained in the comparative tests mentioned in the paper.

Professor Boston calls attention to the fact that in Fig. 19 no information is given with regard to the exact shape of the chips, but only information with regard to their cross-section. I do not recall the exact dimensions of those of my own tests or of those of the other investigators, but in the case of all tests shown in Figs. 19, 20, and 21 t was greater than s.

With regard to the investigations of Dr. Rosenhain to which Professor Hall calls attention, and which, by the way, were published more than a year after my own investigations had first

appeared in print, I fear that a classification of chips as given by Dr. Rosenhain may lead to a misconception as it may suggest that the process of turning is different in one case from what it is in another. Such, however, is not so, as my own observations and motion pictures have definitely shown. The actual process of turning is fundamentally the same, independent of the appearance of the chip. The process of turning, as it appears to me, is always a combination of cutting and tearing.

Mr. DeLeeuw mentions that the tools which he proposed in his paper before the Society in Cincinnati in 1917 were identical with the type of tools I now propose. This statement is based on a misunderstanding. The tools which I have developed and which were used in the tests reported in the last part of my paper, are different with regard to several important points from the tools described in Mr. DeLeeuw's paper. The tool developed by Mr. DeLeeuw signifies only the starting point in the development of the new tool. The depression which I provide is not like the depression produced in time by the chip in the lip surface of a standard tool, but its exact form is the result of very extensive investigations and tests made to determine the best form for efficiency and wearing qualities, taking at the same time into proper account the strength of the cutting edge.

No. 1965

COÖRDINATED OPERATION OF HYDRO AND STEAM PLANTS

SOME OPERATING AND ECONOMIC FEATURES THAT SHOULD RECEIVE CAREFUL ATTENTION

By F. A. Allner[1] Baltimore, Md.

Member of the Society

Due to the low generating costs of modern steam plants the margin of saving offered by a hydro development is frequently quite small. The utmost utilization of the combined hydro-steam source of power has become of greater economic importance in recent years. The economic service of hydro power is usually expressed as replacement value of the equivalent amount of steam power. This comprises two different elements: one, the steam investment cost for the hydro demand service; the other, the steam generating costs for the hydro energy delivered to the load system.

A hydro development that is not equipped with seasonal storage can render the greatest demand service to a given system load if, on the days of greatest steam demand on the system coinciding with minimum flow, the hydro plant supplies the peaks and the steam plant the base of the system load. The greatest energy contribution from the hydro plant is obtained by letting it carry the base portion of the system load during high flow and having the peak portion produced by steam.

General and specific methods are explained by which these economic principles are put into practice. Particulars are given of a system of load signaling by means of small changes in frequency above or below normal which has been found very useful for a more perfect distribution of load during certain hours when it is otherwise difficult to secure the desired division of generation.

Hydro-steam parallel operation may produce a favorable operating combination as these two sources of power can supplement each other to mutual advantage in a number of desirable service features. Hydro units are generally more reliable than steam units. They can be brought up to full load from standstill in a small fraction of the

[1] General Superintendent, Pennsylvania Water and Power Company.

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time that is required for safe operation of steam units. Troubles at hydro stations usually give sufficient forewarning to permit calling in steam assistance to make up for threatened capacity reductions. Inherent limitations which prevent too short a governor traversing time on hydro units are explained, with oscillograms of "load on" and "load off" tests recording speed, gate travel, and pressure in wheel pit. Steam units give better frequency regulation for sudden changes in load. Thus a load system that is served by mixed hydro-steam supply can obtain during a great part of the time certain service advantages of both sources of power without being subjected to the imperfections of either. The hydro-steam combination may at times offer an opportunity of better localizing the disturbing effect of city distribution trouble and of securing a more positive selective relay action than if power were supplied only from one source.

Particulars are given of a proposed ideal hydro-steam parallel operation where the steam station will be located immediately adjacent to the hydro plant. The generating units of both plants will be operated from the same control board. Circulating water for the condensers will be drawn from and returned to the forebay of the hydro plant. During frazil-ice attacks this arrangement is expected to minimize capacity reductions at the hydro plant. There is a natural seasonal diversity in the time of overhauling work at the two types of plants. It will be possible to keep the same skilled labor force employed more steadily throughout the year and to use the same shop facilities for the repair work of both plants.

SUCCESSFUL parallel operation of hydro and steam plants, so far as it refers merely to maintaining synchronism under normal conditions and through the usual abnormal occurrences, presents no particular difficulties comparable with those encountered in some of the earlier problems that had to be solved in the development of long-distance transmission and of distribution networks in large cities. However, in order to obtain the most favorable economic results and the best possible service conditions under certain existing or well-anticipated limitations, a number of seemingly subordinate features should be given careful attention.

2 Very few undeveloped water-power sites within practical transmission distance from the important load centers of the eastern states are so favorably endowed by nature that power from such plants can be delivered to the city networks at a cost very materially lower than that of steam power produced locally at modern large-capacity plants. The efficiency of prime-mover and fuel-burning equipment has shown a distinct upward trend in recent years, accompanied by a general lowering of fuel prices. However, economic comparison with hydro power cannot be set down once for all in definite figures, as the various elements making up the cost of steam power are subject to changes that cannot be

forecast with any degree of accuracy except for a very short span of useful service life. After the hydro plant is completed the total cost for power, consisting largely of fixed charges on the investment, remains fairly constant, although the effective kilowatt and kilowatt-hour service of the hydro plant will show a tendency to enhance in value, particularly that of run-of-river developments operated in conjunction with steam plants on a rapidly growing load system. So to coördinate the parallel operation with the steam plant that it will yield the greatest amount of usefulness, will therefore become of increasing importance until the theoretical optimum is reached.

3 The maximum in energy output will be reached at 100 per cent utilization factor, i.e., when the hydro plant can operate continuously with wide-open turbine gates throughout the excess-flow period. The maximum of peak service can be secured from the hydro plant when the system load has grown to a point where the entire hydro-generating capacity can be utilized on peak generation under minimum-flow conditions, saving then the equivalent amount of effective steam capacity in the carrying of system loads.

4 Broadly speaking, the two sources of power feeding into the same load system make for a rather favorable operating combination. In making this statement it is assumed that the contractual relations under which the two plants operate do not impose any rigid dividing lines as to demand or energy that would be prejudicial to the best utilization of the combined sources of power. This condition does not necessarily require that the two types of plants shall be under joint ownership or operation. It is quite practical to shape the contract relations between the two plants in such a way that fullest coöperation will be encouraged, to nearly the same extent that could be accomplished by joint ownership.

Low-Flow Operation For Maximum Hydro Peak Service

5 In order to derive the greatest saving in investment for steam capacity from parallel operation with a hydro plant, it is necessary, first, to determine the load conditions which will require the greatest steam output. These may not necessarily occur on the day of maximum peak for the year. They will depend on the character of the seasonal variations of system load, on variations in flow as obtained from reliable records covering a long period of years, on the extent of available storage facilities, on seasonal limitations on the effective capacity of a given steam plant due to amount and temperature of circulating water, etc. As a rule it will be found that even for run-of-river developments with only daily or week-end storage, the greatest saving in capacity investment is rendered by the hydro plant on days of maximum annual peak, which should then be assumed to coincide with the day of lowest

winter flow on record. This holds generally true for the load conditions of a system having approximately the same average proportions of lighting, power, and railway load that are now carried by well-developed distribution systems in the large cities of the eastern states.

6 This peak service, measured as a one-hour integration of effective system capacity that can be saved in the steam plants, may be quite considerable, even on a run-of-river development where the minimum winter flow is only a small percentage of the full-draft flow. Accurate advance information on flow and a reliable estimate of the system-load requirements — at least during the peak hours, i.e., the hours of hydro generation — are necessary for actually putting into effect such a division of generation between hydro and steam plants as will yield the greatest hydro peak service.

7 Fig. 1 shows diagrammatically a typical physical layout of a high- and low-tension network supplied from the two sources of power. Fig. 2 shows typical high-flow conditions and Fig. 3 low-flow conditions, both being plotted from actual load data on representative days. The loads are drawn in as one-hour integrations, taken on the hour.

8 For the same year in which the actual load of Fig. 3 occurred, the maximum anticipated winter peak, as estimated at the time when the steam-investment program had to be mapped out, was placed at about 175,000 kw. The lowest hydro energy available during the winter season from the run-of-river development connected to this load system was approximately 300,000 kw-hr., neglecting any use of storage except for the purpose of equalizing the power-house draft during the 24 hours. The peak service obtainable therefrom would have been somewhat in excess of 40,000 kw. The same hydro plant, when fully developed to a delivered capacity of about 80,000 kw., can deliver approximately 2,000,000 kw-hr. at 100 per cent utilization factor; that is, the minimum hydro energy on the critical peak day coinciding with lowest winter flow, is only 15 per cent of the energy ultimately available at high-flow stage. This 15 per cent of energy, however, can be made to render a peak service, measured by one-hour integration, of over 40,000 kw., i.e., nearly 50 per cent of the high-flow peak service of the fully developed hydro plant. As the system load continues to grow at approximately the same load factor, there is of course a tendency for this minimum-flow peak service to gradually increase until it approaches in value the full high-flow peak service.

9 The one-hour integration is not a cŏmplete expression of the effective peak service rendered to the load system, as on the critical peak day the steam plant is assumed to be operated at its maximum sustained output while the hydro plant carries the load variations of the entire system. The maximum indicated load in

excess of the one-hour integration is frequently over 10 per cent above the hourly integration. For the system from which the load data in Figs. 2 and 3 were taken, this excess load was frequently found to be between 15,000 and 20,000 kw. This would raise the effective peak service rendered to the load system by the hydro plant to nearly 75 per cent of the fully developed high-flow peak service.

10 On the critical low-flow peak day the output of the hydro plant, expressed as kilowatt-hours delivered per cubic foot per hour

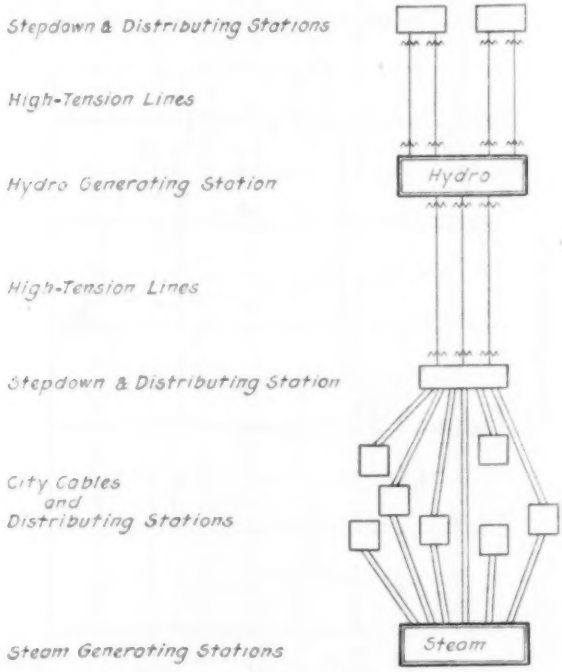

FIG. 1 SYSTEM DIAGRAM

of available river flow, will be slightly reduced by reason of the extremely low-load-factor generation as compared with the delivered output when operating at a high load factor. This slight loss of output, however, is of no moment on the critical low-flow day when maximum peak service is of paramount importance. Even on the average low-flow day this loss of output will usually be compensated for by the greater economy at the steam plant, due to its operation under an exceedingly high load factor. Thus it will be found advantageous to operate most run-of-river hydro plants during the low-flow period in a manner that will give highest

load-factor generation to the steam plant, to which will be assigned the base load. In the example illustrated in Fig. 3 this method of dividing the generation has been carried out even to the extent of ceasing hydro output entirely during the off-peak hours and operating one or two units as synchronous condensers, the power for driving them being supplied by the steam plant.

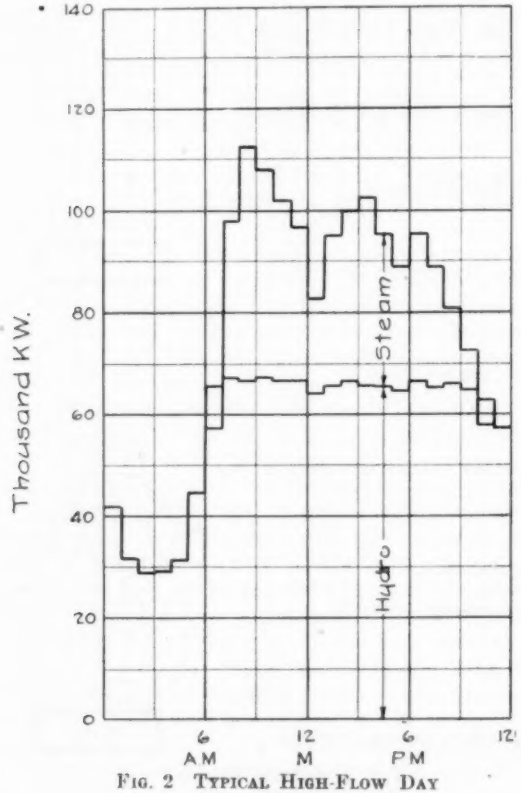

FIG. 2 TYPICAL HIGH-FLOW DAY

(Tuesday, Feb. 15, 1921)

	Kw.	Kw-hr.	Load factor, per cent
Hydro delivered	67,200	1,368,200	84.8
Steam	45,500	442,800	40.5
Total load	112,700	1,811,000	67.4

11 In the case of the typical low-flow day of Fig. 3, there were a number of reasons for this method of off-peak condenser operation: First, the desire to let the steam plant operate even during the off-peak hours at best efficiency, which generally means the loading of steam turbines at near maximum output, and to have

the hydro units come in automatically as spares on unexpected load swings or in the event of failure of a steam unit; second, to assist the steam units in the carrying of wattless current which otherwise might require the paralleling of another steam unit, with a consequent lowering of steam efficiency; third, to better protect the

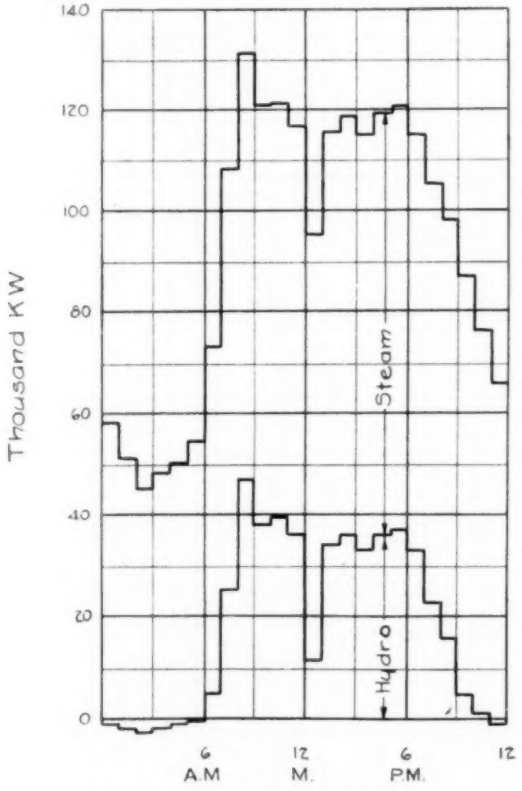

FIG. 3 TYPICAL LOW-FLOW DAY

(Tuesday, Oct. 24, 1922)

	Kw.	Kw-hr.	Load factor, per cent
Hydro delivered	47,700	454,000	39.6
Steam	84,200	1,763,200	87.3
Total load	131,900	2,217,200	70.0

service of the hydro company's other customers as to capacity and also as to better selective relay action in the event of high-tension-line disturbances; fourth, to relieve the steam plant of the voltage-regulating task for the other high-tension-line services, which can

be better taken care of by field adjustments on the hydro generator operating as synchronous condenser.

HIGH-FLOW OPERATION FOR MAXIMUM HYDRO ENERGY OUTPUT

12 During the period of excess flow the hydro plant should be operated for maximum energy output, i.e., with the hydro plant

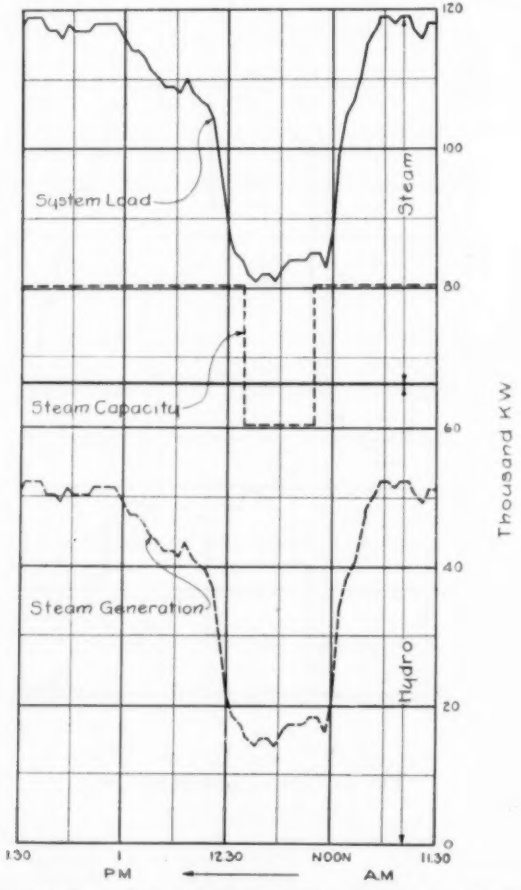

FIG. 4 TYPICAL LOAD DIVISION DURING NOON RECESS IN HIGH FLOW
(Wednesday, Mar. 14, 1923)

carrying the base portion of the system load, and the steam plant the remainder. It will be noted from the load and generating curves of Fig. 2 that the maximum hydro-output idea has not been

carried out to the point of theoretical perfection. For conditions of less than 100 per cent utilization factor at the hydro-plant, there will be at least two — and possibly three or more — periods when an increase of hydro output seems theoretically possible:

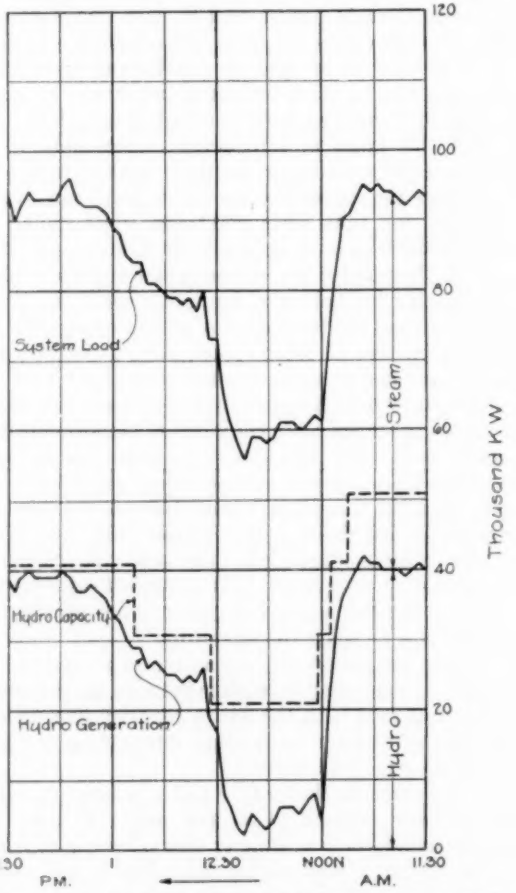

FIG. 5 TYPICAL LOAD DIVISION DURING NOON RECESS IN LOW FLOW
(Tuesday, Oct. 24, 1922)

First, the period around the time when the ascending line of the a.m. load intersects with the hydro capacity line; second, the time of possible intersection of the hydro capacity and load lines during the noon-hour recess, i.e., between noon and 1:00 p.m.; the third period occurs when the descending line of the p.m.

system load intersects with the hydro capacity line; and, depending on relative nearness of the p.m. load curve to the hydro capacity line during certain days or seasons of the year, there may be additional occasions of intersections in the afternoon and evening, before the load remains definitely lower than the hydro capacity.

13 During the first period, that is, near the intersection of the ascending branch of the morning peak with the hydro capacity line, the fires must be brought up rapidly and several steam units must be paralleled in quick succession in order to be sure of carrying the rapidly increasing industrial and railway load. It has been found that this point of intersection varies only within a few minutes from one weekday to another. By a careful following up of the load rise the operator at the steam plant can anticipate with fair accuracy the time when additional units should be started up from standstill so as to be ready for paralleling before the load demands it. Naturally, the steam operator prefers to be on the safe side, having his boilers and prime movers on the line rather a little earlier than too late; he also prefers to have his units carry a certain percentage of rated capacity from the very start, as he wants to avoid just floating the unit in without load and having it motored partly during momentary downward swings of the load, unless the units are specially equipped to stand motoring. Certain types of units at very light load have sometimes less steady governor action than above a certain minimum loading, around 25 per cent. For the foregoing reasons there is a tendency to make a sharp downward dent in the load-dividing line every time a steam unit is being paralleled. The load and capacity changes follow each other so quickly that a perfect adjustment of total load between the two sources of power is not practicable through verbal direction between the operators.

14 During the period of the load recess around the noon hour, an intersection again may take place between the downward branch of the morning load with the hydro capacity line, followed about half an hour or so later by another intersection of the ascending branch of the p.m. load curve.

15 In the example shown in Fig. 4 — a large time-scale reproduction of loads between 11:30 a.m. and 1:30 p.m., taken from two-minute averages of wattmeter charts from a typical high-flow day — the minimum load during the valley of the load curve was high enough not only to load up all hydro units but also give the desired minimum load of 5000 kw. to the steam units on the line. The steam operator in this case disconnected only one of the four units from the bus, as indicated by the dotted line, letting it idle in the meantime on its own governor with reduced auxiliaries.

16 It is interesting to compare the high-flow load division around the noon hour shown in Fig. 4 with a corresponding large-scale illustration drawn from a typical low-flow day (Fig. 5). In the latter case, the steam plant is carrying an almost flat load

except for minor momentary upward and downward swings which the hydro units are unable to take over on account of the longer traversing time of their governors. All gradual and sustained variations in system load, however, are carried by the hydro plant. There were five hydro units on the line, each of about 10,000 kw. capacity, up to 11:53 a.m.; then, within a few minutes' interval, three units were shut down completely until, commencing 12:01 p.m., only two units remained on the line during the extreme valley of the noon drop which generally lasts about half an hour. As hydro units can be started and stopped rather quickly and do not require large power-consuming auxiliaries during the transition, the hydro plant can follow up the noon-hour and other sharp variations of the system load much more promptly with the cutting out and restarting of units than the steam plant, and at a very much lower sacrifice in efficiency.

17 During the third period, i.e., the intersection of the descending branch of the p.m. load curve with the hydro capacity line, the drop in load during the greater part of the year is generally not as rapid as the rise of the a.m. load or the changes during the noon recess. The time of intersection, however, is more apt to shift from one weekday to another, and the operators have to be fully as watchful during that period as on the other occasions to avoid unnecessary loss of hydro output.

18 During the peak season this p.m. drop of the load is somewhat steeper and may at times call for a speedier reduction in the percentage rating of the boilers than is consistent with good maintenance of brickwork and settings. Such a limitation should properly be recognized by the steam and hydro operators, even though it may cause a slight loss of hydro output.

19 The load division during these various load-intersecting periods, as discussed above, was greatly improved on the system from which the illustrations are drawn after a load-signaling scheme by frequency indication had been adopted.[1] Under that scheme the hydro governors were set to maintain approximately one-half of one per cent above normal frequency whenever the plant was not fully loaded, that is, when the hydro governors maintained the frequency. When the hydro plant was wide open, that is, when the steam governors were maintaining the frequency, the latter were so adjusted that they maintained a frequency of half of one per cent below normal. Thus, a glance at the frequency indicator advised the steam operator whether or not the hydro plant was fully loaded. If the frequency was above normal, he adjusted the load on the steam units by switchboard control of the synchronizing springs down to the minimum load desired on the steam unit.

[1] This plan was first suggested in 1919 by F. E. Ricketts, Supt. of Operation, Consolidated Gas, Electric Light and Power Company, Baltimore, Md.

Conversely, the hydro operator could see from the frequency indication, in case it had dropped to normal or slightly below, that his water wheels were wide open and that his governors had ceased to function. If the frequency was slightly above normal, he knew

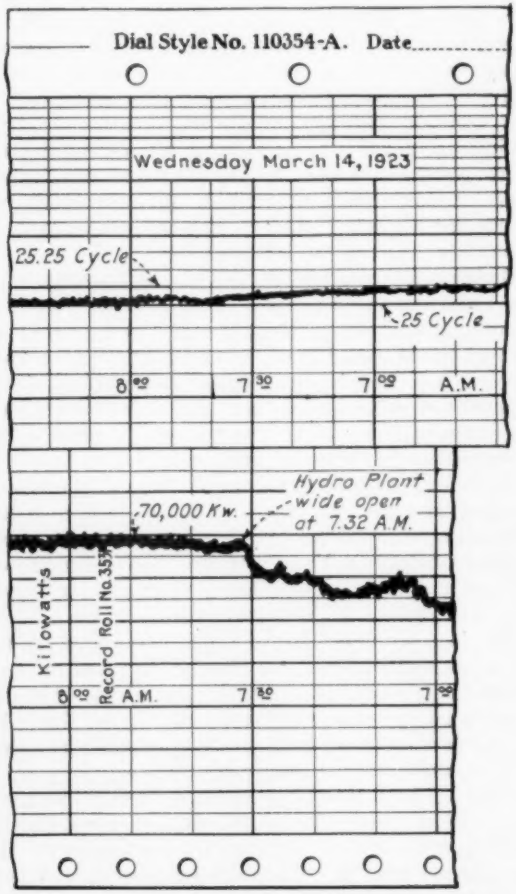

FIG. 6 FREQUENCY AND HYDRO WATTMETER CHARTS, 7:00 A.M. TO 8:00 A.M., ON TYPICAL HIGH-FLOW DAY, ILLUSTRATING LOAD-SIGNALING SYSTEM BY CHANGE IN FREQUENCY

that the hydro plant had still some capacity to spare, and then advised the steam operator from time to time by telephone, after a glance at his station total-output meter, how much capacity he still had in reserve.

20 The practical working of this frequency-signaling plan is illustrated in Fig. 6, which is a reproduction of actual hydro-output and frequency charts during the a.m. intersecting period of a typical high-flow weekday. The frequency is slightly above normal (25.1 cycles) until about 7:30 a.m., and after the hydro plant is wide open (7:32 a.m. on the wattmeter chart), the speed is lowered gradually to 25 cycles.

Service Features of Combined Hydro-Steam Operation

21 A load system that is served by the combination of a hydro and steam plant may obtain under certain favorable combinations a more perfect service than if it were served by either the hydro or the steam plant exclusively. It would be rather difficult to express these service advantages or conveniences in definite terms of money. In many instances they probably are not great enough to influence the initial decision whether to adopt one source of power or the other, which decision will generally be made on economic grounds. Some of these advantages from the system operator's point of view warrant further discussion.

22 Sudden capacity reductions without warning are rather rare at the hydro plant. Statistics recently compiled from an average-service record of large-capacity hydro units showed that the "service demand availability factor" was in excess of 99.5 per cent. This is a measure of the availability of the unit during the time in which it is needed. The fact that on individual units it sometimes is 100 per cent indicates that all maintenance and inspection work can be done while the unit is not in demand. An average taken from the service record of large steam units showed a "service demand availability factor" of approximately 92.5 per cent. This bears out the long-accepted idea that it is not nearly as necessary to provide spare capacity at a hydro plant as it is at the steam plant. It may permit, at times, a substantial saving of investment for spare prime-mover capacity when a system is served by mixed hydro-steam supply.

23 It is not considered safe practice to bring up a large steam unit from standstill to full load in much less than one hour. A hydro unit of nearly equal capacity can be brought up to full load from standstill in less than one minute. Many cases are on record where hydro units were paralleled from standstill in less than half a minute to meet emergencies caused by breakdowns of steam units operating in parallel. By proper training of personnel it has been found possible to make such speedy starting up a regular and safe procedure. Allowing a few more minutes, it can also be made a one-man operation from the control board. Thus, the feeling of security as to carrying of the system load is greatest when one or more hydro units are held in reserve, which is the usual low-flow condition. Even if the hydro plant is shut down during

the off-peak hours in the low-flow period and no units are floated in as synchronous condensers, the hydro source of power can be made quickly available to meet emergencies at the steam plant. Conversely, the steam plant cannot meet a sudden capacity shortage when the hydro plant is wide open, unless some reserve capacity is actually floating on the line.

24 Nearly all the capacity reductions at the hydro plant approach rather gradually permitting sufficient advance warning to call in additional steam capacity early enough to protect the system load. They occur nearly always during the high-flow period when the steam demand is well below the maximum capacity available. These conditions are caused by reduction of head due to rain or ice floods, obstructions of intake screens by trash, and ice trouble in its various forms. Some of the capacity reductions at the steam plant approach gradually, but the great majority come rather suddenly and without warning. To the first group belong shortages of fuel, impairment of feedwater and circulating-water supply, small leakage or coating of condenser tubes, etc. Among the sudden capacity reductions are chiefly breakdowns of the main turbines and generators, although occasionally breakdowns of minor parts of the equipment such as high-pressure piping and boiler-room and turbine auxiliaries may cause enough reduction of capacity without warning to jeopardize the load. If these accidents happen during the hours of wide-open turbine gates in the high-flow period, emergency assistance from the hydro plant will not be available; but, during the entire low-flow period and occasionally also during the off-peak hours in high flow when for some reason or other a small part of the system load is supplied by steam, the hydro plant will generally be able to make up such capacity shortage by speedy paralleling of hydro units, before its disturbing effect is felt by loss in frequency requiring cutting off of load.

25 The speed regulation at times of sudden fluctuations of system loads is somewhat improved by the parallel operation with steam units as compared with all hydro generation. This is essentially due to two causes: First, the kinetic energy per kilowatt of rated capacity that can be absorbed by the rotating masses for a given increase in speed is much greater for the high-speed steam units than for the low- and medium-speed hydro units, even though the latter have generally some extra weight added for that purpose to the electrically active and mechanically necessary materials. The second cause lies in the inherently greater total traversing time of the hydro governor, including therein the time from the first moment of sudden change in load to the moment of reaching maximum change of speed.

26 The traversing time varies somewhat for the same unit, depending on a number of conditions such as the amount of load change, whether thrown on or off, the steadiness of the governor

and relay mechanism immediately before the load change, etc. For the purpose of this comparison the term " traversing time," where not otherwise qualified, refers to the time between the sudden throwing off of 100 per cent load and the moment of reaching maximum speed. At the modern steam turbine this total traversing time can be held down to approximately one second. On certain types of units the standard design and adjustment may give a considerably longer time, around two seconds. Without going into extensive reconstruction work, it is frequently possible to reduce this traversing time by various expedients such as higher setting of the relief valve in the oil-pressure supply, increase of port openings around the pilot valve, large-sized connection pipes to and from the pilot valve, shortening of relay stroke, elimination of all avoidable friction in the governor mechanism and connections, etc. By the shortening of the traversing time on the steam unit an additional service advantage is frequently gained, due to the fact that the overspeed tripping device may not come into play in the event of large loss of load. This greatly shortens the time of restoration of service following an electrical disturbance. Short traversing time on steam governors is particularly desired when operating in parallel with hydro units, because the latter have a tendency to speed up higher for a given percentage of sudden load reduction than the steam units, which, however, being electrically coupled, are pulled along in synchronism. A quicker response of the steam governors will then cut down the percentage load change on the hydro plant as a whole, and will decrease the total amount of overspeed.[1]

27 On the hydro units there are practical as well as inherent limitations on too rapid a governor-relay action, assuming that otherwise it would be possible to make the hydro governor and its mechanism proper fully as free from friction as the steam governors, and also that it would be possible to hold down the lag between movement of the pilot valve and of the relay piston to the same low value as on the steam unit. The latter would be very difficult to accomplish because of the greater serial multiplicity of working parts, the greater length of piping, the much greater inertia of the relay fluids, the greater forces to be handled, etc.

[1] To the author's knowledge, A. L. Penniman, Superintendent, Steam Stations, Consolidated Gas, Electric Light & Power Company, Baltimore, was the first steam-operating engineer who emphasized the great desirability, from the service point of view, of shortening the traversing time of the standard steam-turbine governors. To Mr. Penniman should also be given credit for having originated and actually carried out on standard governors of certain types of steam turbines most of the above-mentioned expediencies, which rendered them more suitable for perfect hydro-steam parallel operation.

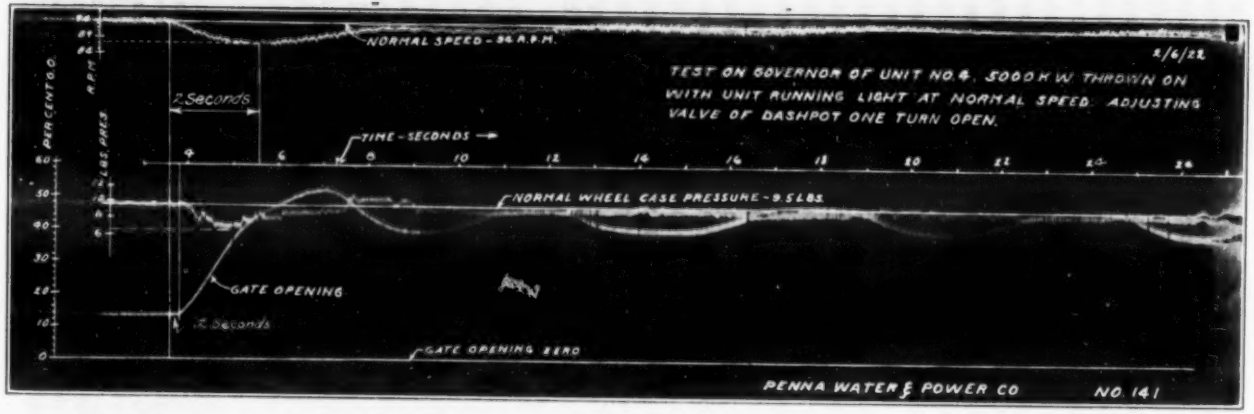
NORMAL SPEED - 96 R.P.M.
2 Seconds
TIME - SECONDS
TEST ON GOVERNOR OF UNIT NO. 4. 5000 K.W. THROWN ON WITH UNIT RUNNING LIGHT AT NORMAL SPEED. ADJUSTING VALVE OF DASHPOT ONE TURN OPEN.
2/6/22
NORMAL WHEEL CASE PRESSURE - 9.5 LBS.
GATE OPENING
2 Seconds
GATE OPENING ZERO
PENNA WATER & POWER CO
NO. 141

FIG. 7 DIAGRAM OF SPEED, GATE OPENING, AND WHEEL-PIT PRESSURE OF 10,000-KW. HYDRO UNIT, 53-FT. HEAD, FOLLOWING SUDDEN THROWING ON OF 50 PER CENT LOAD

28 Sudden reduction of water flow causes a pressure rise in the wheel casing, penstocks, etc., the water hammer of which may not only stress these parts to the breaking point but may also, on sudden gate movements, nullify the effect of reduced passages. That is, the discharge through the smaller openings at the increased pressure may impart greater power to the wheel than the discharge through the previous larger openings at normal head. This condition is illustrated in the oscillograms of Figs. 7 and 8, where wheel-pit pressure, speed, and gate travel are recorded from a 50 per cent "load on" and "load off" test, made on a 10,000-kw. hydro unit operating at 53-ft. head. The effect of the rise or drop in water pressure is not completely recorded there, because in addition to the pres-

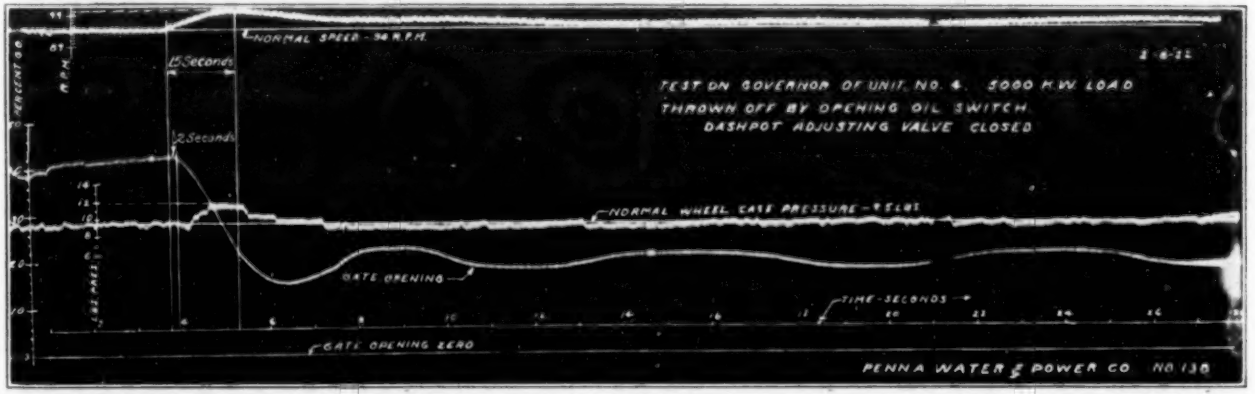
NORMAL SPEED - 50 R.P.M.
1.5 Seconds
2.5 Seconds
TEST ON GOVERNOR OF UNIT NO. 4. 5000 K.W. LOAD
THROWN OFF BY OPENING OIL SWITCH.
DASHPOT ADJUSTING VALVE CLOSED
NORMAL WHEEL CASE PRESSURE - 9 LBS.
GATE OPENING
TIME - SECONDS
GATE OPENING ZERO
PENN A WATER & POWER CO NO 138

FIG. 8 DIAGRAM OF SPEED, GATE OPENING, AND WHEEL-PIT PRESSURE OF 10,000-KW. HYDRO UNIT, 53-FT. HEAD, FOLLOWING SUDDEN THROWING OFF OF 50 PER CENT LOAD

sure changes above the wheel there are corresponding changes taking place in the draft-tube effect below the wheel, both of which work in the same direction, i.e., they impede the power-regulating effort of the governor.

29 The traversing time at 50 per cent load change was found to be between 1.5 and 2.0 sec. For the 100 per cent "load off" test the shortest traversing time measured on the same unit was 2.6 sec. The higher the head at the hydro plant, the greater the traversing time necessary for handling the same amount of water on a unit. Minimum values for traversing time are frequently 6 sec. and greater in plants operating at heads of 200 ft. or higher.

30 On a well-laid-out high-tension transmission system, lightning has ceased to be a serious cause of more than

momentary capacity reductions, and even the voltage-disturbing effect on the city distribution system can be greatly minimized by proper sectionalizing of buses and quick-acting selective-relay devices. On a typical layout such as shown in Fig. 1, the hydro-steam combination may offer an opportunity of better maintaining bus voltage on parts of the city system in case of cable breakdowns or of trouble at distributing stations, thus localizing the disturbing effect on the commercial load as a whole.

Effect of Hydro-Steam Parallel Operation on the Average Cost of Steam Generation

31 The question is frequently asked whether a steam plant, when operating in parallel with a hydro plant in supplying a given system load, could be expected to show a lower operating and maintenance cost per kilowatt-hour than if the system load were supplied exclusively by the steam plant. There are no actual cost figures available from any installations of appreciable size, say, 50,000 kw. or over, which cover identical conditions for a sufficient length of time to permit setting up reliable cost figures that are strictly comparable for the two operating conditions.

32 There is little doubt that the average steam cost per kw-hr. would be higher in the case of mixed supply when the hydro plant is an all-year-round steady source of power, supplying the base portion of the system load. Inversely, the kw-hr. cost of the steam portion of the mixed supply will assuredly be lower than for all-steam supply in case a hydro plant that is equipped with seasonal storage furnishes the peak portion of the system load through the greater part of the year.

33 In the case of run-of-river developments, hydro-steam parallel operation, if conducted under a coöperative plan, will show a lower average steam cost during low flow but a higher cost per kw-hr. of steam generation during high flow. The overall result will probably be controlled by the weighted preponderance of the higher or lower load-factor generation compared with the load factor of the total system load. A steam station operating in parallel with a run-of-river hydro plant is subject to considerable variation of operating conditions in the course of the year and, at times, also to sudden changes from one day to the next. In order to obtain the best possible results under these conditions from the combined sources of power, the personnel at the steam plant has a more painstaking task to perform than in those stations where the entire system load is supplied by steam or where the hydro plant furnishes a fixed portion of the load throughout the year. Sudden changes from large-output generation at the steam plant to reduced output are much more frequent than the sudden changes from small to large generation. This is due to the fact that a low-river stage may at times develop over night into a high-flow stage, but the drop in flow will always be more gradual.

Seasonal Diversity in the Time of Maintenance Work for Hydro and Steam Plants

34 If the regular seasonal high-flow period occurs with a fair degree of certainty for several consecutive months each year, the mixed supply will offer to the steam plant an opportunity of carrying on major repair work and routine overhauling in an efficient prearranged manner. This should have a tendency to not only lower the unit cost of maintenance per kw-hr. but should also make possible certain savings of investment in spare parts or in whole spare units that otherwise would become necessary under all-steam supply to compensate for the capacity withdrawals during the period of overhauling.

35 A still further going advantage in the conduct of maintenance work can be realized when the hydro and steam plants are located in close enough proximity to permit the utilization of the same maintenance force and, partly also, of joint shop facilities for the routine overhauling work of both plants. Such increase in load factor on repair equipment and skilled labor is favored by the natural diversity in the time of repair work for the two plants. The run-of-river hydro plant will always endeavor to have all its equipment in continuous service during the high-flow season. Therefore little opportunity will be had then to take equipment out of service except for the rare occasions of necessary repairs of breakdowns. The high-flow period, on the other hand, means generally an easing off of capacity demand on the steam plant. Major steam-maintenance work that requires a protracted withdrawal of steam equipment can therefore be best attended to during that period.

36 Again, during the low-flow season the combined maintenance force can be chiefly engaged on the work at the hydro plant, as this is the time when all available steam equipment may be called on for maximum output, while hydro equipment can be taken out of service without loss of output. The low-flow season is therefore the best time to carry out all the major repair work at the hydro plant and all of the preventive maintenance work. The latter is a very important feature of good hydro operation, as it will then better assure freedom from hydro breakdowns during the high-flow season.

Ideal Conditions for Combined Hydro-Steam Operation When Steam Plant is Immediately Adjacent to Hydro Station

37 A steam station immediately adjacent to a run-of-river hydro plant of 150,000 hp. rated capacity is now under construction in southeastern Pennsylvania. The initial installation will consist of two 12,500-kva. steam turbines. These first two and all the future generating units will be connected directly to the low-tension bus of the adjoining hydro station and will be operated

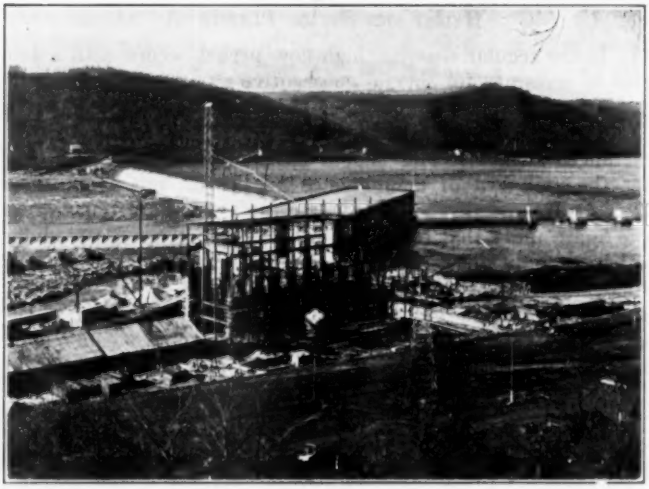

FIG. 9

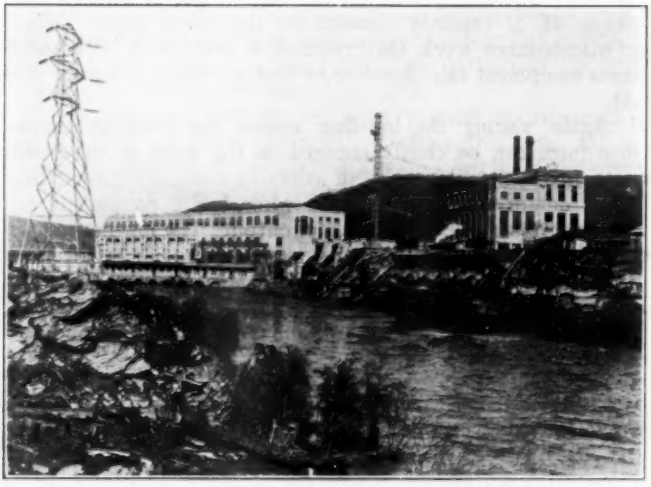

FIG. 10

FIGS. 9 AND 10 VIEWS OF HYDRO PLANT WITH ADJACENT STEAM STATION UNDER CONSTRUCTION, TO BE CONTROLLED FROM SAME SWITCHBOARD.

from the same control board. An unusual opportunity will be had at this installation to utilize the previously mentioned diversity in the time of steam and hydro maintenance work. From the steam point of view a rather favorable layout is offered at that station (Figs. 9 and 10) for the arrangement of conduits for the circulating water, which is drawn from the forebay of the hydro plant and returned there. This is expected to reduce materially the danger from frazil-ice attacks, at least to that group of hydro units where the intakes are located near the outlet of the discharge conduits.

38 Bearing on the subject of hydro-steam parallel operation, an effort has been made at this plant to obtain for the first two units the shortest possible starting time, to permit their being floated in at no load and also to permit motoring from the hydro plant for short periods in every-day high-flow operation. The governors are designed for very short traversing time and will permit adjustment of regulation and of range from the control board while running. By this arrangement a better control will be obtained for any desired division of load as between hydro and steam units and also in regard to the steam units at other stations. These features will greatly assist in obtaining the maximum theoretically possible hydro output during the high-flow period.

39 In this particular installation a number of other economic and service features will be carried out, which will tend toward the best utilization of the combined sources of power. On account of the close proximity and joint control of the two stations it will be possible in this new type of plant to put into practice the ideal of hydro-steam parallel operation to a much greater extent than is feasible under the usual conditions of mixed hydro-steam supply.

DISCUSSION

A. G. CHRISTIE.[1] This excellent paper indicates in a clear manner the relations that exist between the steam and hydro plants of at least one group of interconnected plants. No doubt similar conditions exist elsewhere and the same care may have been taken to study the most economical use of the two types of plant. The average engineer, however, is not always familiar with these facts. The general public does not realize the limitations of low flow in run-of-river plants nor does it know that heavy expenditures must be made to provide steam-generated power during low-flow periods. This increased first cost with its fixed charges must be reflected in the rates charged for service. Politicians and demagogues have led people to believe that water power is cheap power; and there is apt to be an unfortunate reaction when the public realizes that this cheap power cannot be obtained. In this con-

[1] Professor of Mechanical Engineering, The Johns Hopkins University, Baltimore, Md. Mem. A.S.M.E.

nection, particular attention and publicity should be given to the author's second paragraph where he says:

> **Very few undeveloped water-power sites within practical transmission distance from the important load centers of the eastern states are so favorably endowed by nature that power from such plants can be delivered to the city networks at a cost very materially lower than that of steam power produced locally at modern large-capacity plants.**

It will be apparent from a perusal of this paper that steam stations connected to the system are of increasing importance for economical use of the water power as the system load grows. The data presented would indicate that the water power at Holtwood is being developed economically within the limitations of total system load. Further growth of load will make its more complete economic use possible and will also call for additional steam capacity, some of which will be furnished by the steam plant now under construction at Holtwood.

In steam stations of this character, it is a very difficult matter to make an estimate of the probable use factor for the steam equipment on which to base the economic selection of plant machinery. The writer has had to study several such problems at both the Baltimore and Holtwood stations. Other problems in steam-plant design have been mentioned incidentally by the author. For instance, during periods of high-flow, the steam station must pick up practically its whole load each day from a dead condition within a period of about an hour. This is severe service. What types of turbines, boilers, furnace equipment, and auxiliaries are best suited for this service? The problem in another hydroelectric system that the writer is studying is totally different. Its steam station for many years will be a stand-by for a relatively few hours' operation per year of rather large-capacity peak-load service. No base-load service can be expected on the equipment first installed. It will be evident that few steam-station refinements can be warranted in such a plant as compared with one operating under Baltimore conditions. Engineers would be interested in figures on the use factors of the steam stations in Baltimore and of the hydro plant at Holtwood if the author can give these. Such figures are tentative only, for low flows vary in period and amount from year to year.

The new steam station at Holtwood, briefly discussed by the author, has several features of special interest to engineers. It is hoped that full details of this plant will be submitted after it has been put in operation and experience gained with the equipment that has been installed.

CHARLES B. HAWLEY.[1] In discussing the operation of interconnected hydro and steam power plants, the author has called

[1] Consulting Engineer, Washington, D. C.

attention to the fact that best overall economy will result when it is possible to carry the system peak loads on the hydro plants and the base loads on the steam plants. In order to take advantage of this ideal method of operation throughout the entire year and avoid the duplication of hydro installation by reserve steam capacity, seasonal storage is required for the water powers.

It is seldom that a single water-power site can be economically developed so that it will have complete seasonal regulation. Usually the cost of providing large storage is so great that it must be distributed over two or more installations on the same river system in order that the expense may be justified. Likewise, a single power development on an unregulated stream, if charged with the cost, maintenance, and operation of the necessary steam reserve, will rarely be able to improve by a very wide margin on the operating costs of large modern steam plants. For greatest

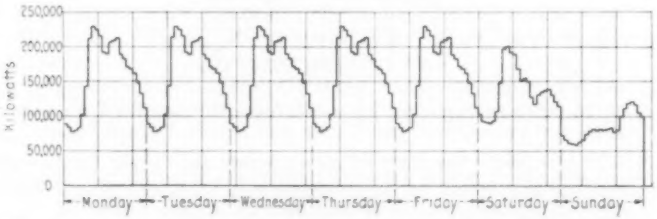

FIG. 11 TYPICAL WEEKLY LOAD CURVE OF A LARGE DISTRIBUTION SYSTEM

Power generated during week: Monday to Friday, inclusive, 19,145,500 kw-hr.; Saturday, 3,285,400 kw-hr.; Sunday, 2,019,100 kw-hr.; total power for average week, 24,450,000 kw-hr.; average power for week, 145,535 kw.; average weekly load factor, 63.1 per cent.

economy, therefore — and these are the points the writer wishes to emphasize — a river should be developed as a whole with seasonal storage, located, so far as is practicable, at the headwater plants, and preferably at medium- or high-head installations.

The writer also wishes to show graphically the effective service which hydro plants can render to steam power plants by carrying their peak loads. The load diagram in Fig. 11 is that of a large interconnected power system and indicates typical variation in demand over the period of one week. The curves in Fig. 12 are derived from this load diagram and show the exact amount of steam plant capacity which can be replaced by any given amount of primary water power, it being assumed that the hydro plant has sufficient installation to permit operation at low-load factor.

The maximum demand for one hour is 230,000 kw.; the instantaneous demand will of course be greater. A hydro plant of 50,000 kw. primary water supply and 150,000 kw. of generating capacity will take all the peak load of this system and permit a

100,000-kw. steam plant to operate at from 90 to 100 per cent load factor on the base load.

While it would be difficult to maintain such an ideal distribution of load between interconnected hydro and steam plants owing to the continual increase in system demand, and the intermittent

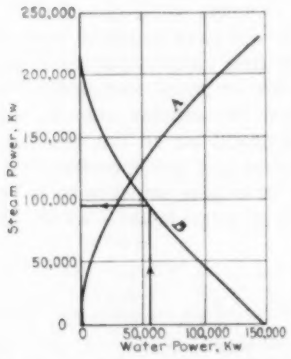

FIG. 12 CURVES SHOWING ADVANTAGEOUS DISTRIBUTION OF LOAD BETWEEN INTERCONNECTED HYDRO AND STEAM POWER PLANTS

A—This curve shows how much the power demand (peak load) on the steam system can be reduced by using any assumed amount of primary hydroelectric power; *B*—This curve shows how much primary hydroelectric power is required to reduce the power demand (peak load) on the steam system to any assumed amount.

addition of power capacity, yet the general principle is sound and will serve as a guide in proportioning the capacities of hydroelectric developments.

N. E. FUNK.[1] The writer desires to emphasize a point in the paper, because many engineers, in making comparisons, have a tendency to take figures from papers without giving sufficient regard to conditions under which those figures were secured.

Figs. 2 and 3 show that certain load factors can be developed by the hydro plant on base load, and that certain peaks can be developed by the hydro plant on peak load. If the system load were doubled, the load factor on the hydro plant would be increased from 84.8 per cent to 98 per cent, and the peak would be increased from 47,700 kw. to 65,000 kw. Practically a full-capacity peak use, and almost 100 per cent of the water at base load can be obtained, with twice the load to absorb the water power. This shows that figures of utilization of existing water-power installations cannot be taken offhand and compared with proposed installations under different conditions. This is done in many cases,

[1] Operating Engineer, The Philadelphia Electric Company, Philadelphia, Pa. Mem. A.S.M.E.

without giving due regard to the ability to absorb the power during the two different times of operation, base and peak. The same number of kilowatt-hours can be concentrated in a greater peak for a shorter time. For instance, 100,000 kw. for two hours is only 200,000 kilowatt-hours. If the load absorbing the water power were half the value it would be necessary to run four hours for the same energy output. This would mean getting down to the wider part of the load curve and really necessitate running more than four hours, which would mean more than 200,000 kw-hr., and as there might not be that much water available, the peak would be reduced to less than 50,000 kw.

C. W. PLACE.[1] In 1921 the writer presented a paper[2] before the Society covering in general the type of operation that the author has described in his paper. The latter has brought out with operating facts at hand a number of points which the writer attempted to make, especially regarding the peak operation of hydroelectric stations, and among these may be emphasized the following:

He has based his arguments upon results obtained from large concentrated loads, supplied by hydroelectric and steam-driven generator stations.

It would seem that the importance of peak service by hydroelectric stations to lighter-weight systems, which cover a larger area, would be even greater than described. This would be particularly true where there is a steam base-load plant near the principal concentration of load and the hydroelectric stations are located throughout the transmission district surrounding this load concentration. The transmission and transformation losses, under these conditions, would enter in as credit to this hydroelectric peak-load service as well as the effect of improving the load factor on the steam equipment.

The paper leads to the inquiry as to whether the proper way to think of hydroelectric power is not in terms of peak power and off-peak power, rather than as primary and secondary, or firm and dump power. The true value to a system is due to the improvement in load factor on the steam equipment and consequent decrease in cost of the total steam production, rather than in reducing the coal consumed in such steam production at perhaps inconvenient times.

The comments on slight changes of frequency to indicate load conditions are interesting. In 1919 we put in a station which was brought on automatically by a droop of frequency, i.e., on an indication of excessive load in comparison to generating capacity, and which was taken off the system on loss of load. This was

[1] General Electric Company, Chicago, Ill.
[2] Trans. A.S.M.E., vol. 43 (1921), p. 437.

carrying this indicating scheme a little further. It meant, however, that capacity to carry load would be quickly available, either as the peak came on or as something happened to the generating equipment in operation.

Another factor allied to the writer's discussion is that there are few hydraulic plants installed which cannot do something along the line of operation described in the paper. There are very few with ponds so small that if the pond is pulled down a little during the peak it will not allow a greatly reduced generator capacity during the off-peak period without wasting water over the dam.

This is true even in rivers like the Wisconsin with its series of developments, and is due to the increase of interconnections and the exchange of power possible between different interests, and will become more effective as further interconnections are made.

JOHN F. VAUGHAN.[1] The author's paper, which is very pertinent at this time, suggests two vital points. First, in the cost of power, in the case of water, the proportion in fixed charges (cost of money) is relatively large and the proportion of variable charges (cost of labor and materials) relatively small, while with steam the fixed charge is smaller and the variable operating cost relatively larger. On the other hand, there is a greater margin for gain in efficiency(reduction of losses in the conversion of energy) in the case of steam. That is, with water there is greater stability in the cost of power and less uncertainty in forecasting it, while with steam there is less certainty but perhaps more opportunity for improvement in methods to offset a future continuance of rise in labor and material.

In the second place, the position of the local source of power, whether water or steam, in a market supplied with power brought in over long distances from more favorable sources, is of equal interest. The cost of transmission over long distances is principally fixed charges, i.e., the cost per kilowatt of capacity is fairly rigid. The unit cost (per kw-hr.), on the other hand, is dependent on the amount of energy transmitted. With a rigid numerator (total cost of transmission), the denominator (kw-hr.) must be increased to produce a reasonable quotient (cost per kw-hr.). The prime requisite, then, is high load factor. In so far as the local source of power can economically provide a kilowatt of capacity for the relief of peaks of load, or can fill in an area of kilowatt-hours of energy during longer periods of heavy load, such a power takes on an enhanced value.

W. W. EBERHARDT.[2] The operating conditions on the Alabama Power Company's system are very similar to those on the Pennsylvania Water Power Company's system, as described by the

[1] Vaughan Engineers, Boston, Mass. Mem. A.S.M.E.
[2] Operating Department, Alabama Power Co., Birmingham, Ala.

author. The installed hydro capacity, which is made up entirely of run-of-river developments, is 145,000 kw., and the steam capacity is approximately 90,000 kw., exclusive of the 60,000-kw. Muscle Shoals steam plant which the company has under lease from the United States Government.

During the wet season, or the months from January to April, inclusive, the steam stations are called upon only for peak-load service and carry that portion of the system load above 145,000 kw. During the extreme dry season, or the months of September to November, inclusive, the available hydro energy is reduced to 500,000 kw-hr. per day, making it necessary to limit hydro-plant operations to the peak-load hours of the day.

If, during the low-flow period, the steam plants are given a straight base load and the hydro plants are made to carry the load fluctuations, the efficiency of the hydro plants will be greatly reduced, due to the resultant low-load-factor generation. It has been found in our case, that by causing the steam plants to take about 5000 to 10,000 kw. of the load swing, a considerable saving in system efficiency can be obtained because the hydro units can then be operated at their most efficient gate opening at all times. In the morning when the hydro units are brought on the line to pick up load they are given a good load as soon as possible, reducing steam generation if necessary.

When the load is dropping off the same procedure is followed, that is, hydro units are kept loaded until taken off the line or brought to a floating operation with vacuum broken in draft tubes and generators operated as synchronous condensers.

In general, it will be found that steam units can carry light loads with better efficiency than the hydro units. The maintenance at the steam plants, of course, is increased, but is more than compensated for in the saving in maintenance at hydro plants. In some hydro plants it may be necessary to operate units at good load, regardless of whether any gain in efficiency is obtained, in order to avoid pitting of water-wheel runners at light loads.

In Par. 19 the author describes a very ingenious method of controlling the load division between hydro and steam stations by means of frequency indication. This method, as stated, requires a frequency variation from 0.5 per cent above normal when the hydro station carries the swing, to 0.5 per cent below normal when the steam station carries the swing, or a total variation of 1.0 per cent over the extreme operating range. It is interesting to note that the Pennsylvania Water Power Company's customers will tolerate such a frequent variation. On the Alabama Power Company's system and other southern systems with large cotton-mill loads, such a frequency variation would not be tolerated, because of the resultant variation in the speed of the looms.

At any rate, it seems that such a scheme of control by frequency indication is limited to a simple system having only a few gen-

erating stations as the author describes. When several systems are interconnected as in the Southeast, where as many as 16 hydro and 12 steam stations may be operating in parallel, the division of load by frequency indication, even if it would be tolerated, could not be applied because of the different operating conditions on the various systems. That is, a hydro station on one system may be carrying the load swings, whereas on another system a steam station may be carrying the swings. Furthermore, with a variable frequency it is extremely difficult to control the power flowing over the tie lines between the interconnected systems. When the systems are interconnected it is generally desirable to control the flow of power from one system to another. On the Gadsden-Lindale line which connects the Alabama Power Company and the Georgia Railway & Power Company systems, the power flow at times has been observed to reverse in direction on a change of frequency.

The control of power flow by the load despatchers has been found to be very difficult, because the telephone communication requires several minutes, during which time there may occur a radical change in conditions. It has been found that the only satisfactory method to control the power flow between systems is to rigidly maintain the frequency at its normal value. To this end all governors on both hydro and steam units must be properly adjusted and very accurate frequency-indicating devices must be installed in the frequency-controlling station on each system. As a result of this method of operation there are times, of course, when losses will occur in hydro generation, especially during the morning hours when load is picking up and during the noon-hour period of the high-flow season.

On the Alabama Power Company's system the principal steam plant is a mouth-of-mine station located on the opposite side of the load center from the hydro station. In addition to the obvious advantage of locating a steam plant near the coal supply, the connection of steam and hydro plants to the load center by separate transmission lines has worked out to advantage in the maintenance of service and the effective use of idle generating units as synchronous condensers, that is, spare units are available at the steam plant in the high-flow season and at the hydro plant in the low-flow season, for condenser operation. At the Gorgas Steam Plant, one 25,000-kva. unit is especially equipped with a coupling so that the generator can be disconnected from the turbine and operated as a synchronous condenser, thereby obviating the necessity of feeding steam in the turbine for cooling purposes. By having synchronous condensers on the opposite side of the load center from the base generating station, the same effect, as far as transmission-line regulation on the loaded line is concerned, is obtained as if the condensers were actually at load centers, that is, the lines between the load center and generating station supplying the base

load are relieved of all wattless load and can consequently be operated at unity power factor with a resultant increase in capacity.

WM. W. TEFFT.[1] We have 37 hydro plants on the system of the Consumers Power Company interconnected by means of transmission lines with 12 steam plants.

In the early development of the system the matter of governors on these developments was of great importance, and their sensitiveness was designed to be rather high in order to maintain close speed regulation. At that time steam plants were only secondary, and even though their regulation sensitiveness was greater than for the hydraulic plants, they were not large enough to lift the load even though they went into it quicker than the hydraulic units. On this account it was vital that the hydraulic units pick up their share of the load as quickly as might be and so the wr^2 was specified as a high amount and the sensitiveness of the governors emphasized.

Now that the system has grown, and the steam capacity has become great enough to pick up the momentary fluctuations of load, the importance of the sensitiveness of the hydraulic governors has practically disappeared. Their main use now is to keep the unit from running away in case it goes out of step, and to bring the unit up to synchronous speed. It is our experience that if the governors on any one hydraulic plant are set to govern closely it will cause the governors to hunt, because, if the frequency falls off slightly, the governor opens wide and the capacity of the plant is not great enough to lift the frequency and so it stays wide open until some steam plant does lift the frequency a little, and then the hydraulic governor will close entirely or nearly so, as the subtraction of all of its power is not enough to lower the frequency of the system. Our general operating policy is to have the steam plants take the base load and the hydraulic take the *general* fluctuations of load. For the small momentary fluctuations of load the steam plants handle the governing. The writer doubts if any system anywhere has better frequency regulation than the Consumers Power Company. It is so good, in fact, that knitting, textile, and paper mills are not at all disturbed by fluctuations of frequency.

This, then, leads to the consideration of gate opening on the turbines. About four years ago this matter was very thoroughly worked out on this system and the method adopted of operating hydro units at the best gate opening *only*. This is made possible by the great size of the system and the multiplicity of hydro units on it, so that if additional load was required it might be

[1] Vice-President and Chief Engineer, Consumers Power Company, Jackson, Mich.

obtained by placing an additional unit on the line somewhere at close to its best gate opening, and all the other units that were already on the line could be kept at their best gate openings. This resulted, as nearly as could be determined, in an increase in kilowatt-hours per annum of somewhat over 11 per cent. This was of course a very definite benefit, and the amount of it was also definite.

In our new plants, the governors are being omitted for the most part and automatic operation being adopted. Automatic operation of course operates the unit at best gate opening or else shuts it down. In making an economic comparison of automatic operation with manual operation we do not give the automatic operation the benefit of this 11 per cent, because now we are operating our governor plants on the same gate opening.

At our Foote Dam on the Au Sable River the draft tubes were aerated some years ago so that the generators could be used as synchronous condensers, and thus avoid the loss of power necessary to drive the turbines as pumps in the draft-tube water. This method worked out to our entire satisfaction and only required that the steady bearings be lubricated whether the water was in the draft tube or not.

The importance of synchronous condensers on this system has decreased somewhat since then on account of the larger amount of power that is always coming down the Au Sable transmission line, but for short periods when very little power was required the synchronous condensers were of great value in energizing the line and thus avoiding the necessity of operating all three units at Foote, such as at 0.3 gate, for the sole purpose of obtaining capacity effect for the line.

It appears to have originated with this company to lap the heads in a series of dams. Space cannot be taken here to discuss this, but in general the headwater of the downstream dam is raised until it laps the tailwater of the next dam upstream by 3 or 4 or 5 ft., as the topography allows. This has the effect of creating pondage without additional evaporation and without loss of head, as whatever head is subtracted from the downstream dam is added to the upstream dam, and it also corrects a very serious loss of power. Most power companies must operate their dams through 12 hr. out of each 24 hr., which has the result (if they use all of the water which comes down the river) of lowering the headwater for 12 hr. and raising the tailwater for the same 12 hr. from normal, thus decreasing the effective operating head. A careful investigation of the Mio dam on the Au Sable River for one year shows that the loss due to this cause was slightly over 8 per cent of the power that came down the river. This is entirely corrected, except, perhaps, for the loss of 0.2 or 0.3 ft. of head, by the lapping of heads.

E. B. POWELL.[1] The writer was especially interested in the outline of the principles of steam-plant and hydro-plant parallel operation under conditions of variable water supply, utilizing the peak capacity of the hydro plant under one condition and its maximum energy-generating capacity under the other. We began that general method of operation about 1908 when we undertook the scheduling of operation for our different plants. Since then, of course, methods have been elaborated upon.

On reading this part of the paper it occurred to the writer that even greater emphasis might be placed upon the necessity for careful economic balance in scheduling the relative operation of the two types of plant. As our power systems grow larger and small differences in efficiency percentage run to large amounts in dollars and cents, scheduling operation becomes a matter of close attention to detail. The work is simplified if the efficiencies of the various primary energy supplies are first separately evaluated in terms of a common unit and their individual characteristics as affected by load, flow, head, and other conditions presented on this basis. The operation of the several parts of the system may then be so scheduled that the total overall expenditure for energy shall be a minimum. As brought out in the paper, such scheduling of joint operation of hydro and steam plants requires a thorough advance knowledge of stream flow and load characteristics. The writer believes, however, that this scheduling should not be regarded as other than a normal routine requirement on a large or important power system.

THE AUTHOR. Professor Christie's inquiry about the use factor of the hydro plant in Holtwood and of the steam station in Baltimore, invites more than merely a brief answer in figures.

For the three years, 1921 to 1923, the average use factor at the Holtwood plant was about 55 per cent and at the Westport steam station about 25 per cent. It must be considered, however, that the hydro plant operated, so to speak, without spare prime-mover capacity, while the steam plant had considerable spare prime-mover capacity, some of which was not operated at all. Preference was given at the steam plant to the more efficient prime-mover capacity, and the use factor on this part of the equipment was comparatively high, comparing favorably for individual units with base-load conditions in the average steam station. The load factor of the total steam generation at the Westport plant, obtained as weighted daily average, was approximately 70 per cent during the three-year period.

Mr. Hawley expresses himself very strongly in favor of seasonal storage as being required for the ideal of coördinated hydro-steam

[1] Consulting Engineer, Stone & Webster, Inc., Boston, Mass. Mem. A.S.M.E.

operation. It is obvious that for a given load system and a given annual hydro output the value per kilowatt-hour of hydro output, expressed in terms of steam power and energy replaced, will be greater when there is complete seasonal storage than when the hydro plant is equipped only with partial or no seasonal storage. There are hydro developments that are best suited as storage projects, certain others that are feasible only as run-of-river plants, and there is a third kind of rather exceptional occurrence, i.e., the nearly constant-discharge plants (Niagara Falls, St. Lawrence). Practically all of these three types of plants will be found to operate in conjunction with steam plants, but the ideal economic coördination of the hydro and steam sources of power will require peak operation throughout the year for the storage type of plant, and base-load operation for the constant-discharge plant. A place in between will be occupied by the average type of run-of-river plant; during the excess-flow period its every-day operation will approach that of the constant-discharge plant, but during the low-flow period it will resemble that of a seasonal-storage plant.

Seasonal storage must usually be figured as an additional investment in excess of the daily or week-end plant storage, which is inherent to most hydro projects where a good part of the operating head is created by a dam construction. It can readily be seen from the charts used by Mr. Hawley that the first stages of such additional storage, expressed in units of acre-feet or of annual energy, will be more valuable as replacing steam investment and low-load-factor steam generation than any succeeding stages of flow equalization. Frequently a limit will be found beyond which provision of additional seasonal storage will not be found economically attractive. This applies particularly to a plant or a group of plants where the increment investment per kilowatt capacity delivered at the load center, i.e., cost of hydro capacity plus transmission and low-tension tie-in equipment, is greater than the increment cost per kilowatt of steam capacity.

Mr. Hawley's method of determining the best division between hydro and steam generating capacity is well adapted to those developments where there is a great latitude in the amount of seasonal storage that can be provided. This enables the designer of the hydro plant to make provision in advance for a flexible coördination of the two sources of power, in investment as well as in the conduct of the operation.

The type of study suggested by Mr. Hawley will also facilitate the holding down of the initial step of the hydro investment to a size that will insure economic success. Subsequent steps, either in additional plant capacity, or in storage, or possibly also development of successive stages of head on the same river, can be added selectively in future years, timing the period of best usefulness on the growing load system with that of attractive return on the increment investment.

Mr. Funk selected a good example of demonstrating for a given hydro plant a greatly increased value of energy output (84.8 to 98 per cent load factor on high-flow weekdays) and a proportionately still greater gain in peak service (47,000 kw. to 65,000 kw.) by simply doubling the scale of the system load curve. Such a doubling of the load may not be far off from actual conditions met in the history of a hydro plant. It may be due to the natural increase of the originally connected load system; or it may be due to the tying-in of additional load groups or due to a combination of the two conditions. The tying-in of two or more systems of well-known load characteristics furnishes a more reliable basis for figuring the increased power value from a given hydro plant than merely estimating the growth of load of the same power system for a long period of years, assuming the same shape of load curve.

Practically all distributing companies in the large cities have experienced in the past and are still striving further for an improvement in load factor, which manifests itself ultimately in the shape of the load curve in two ways: first the filling in of the off-peak valleys (midnight to 6:00 a.m. on weekdays and the whole day of Sunday); and second, in the broadening of the sharp peaks at day time. These sharp peaks are now chiefly caused by the overlapping of lighting and traction load while the industrial load itself is relatively free from pronounced peaks. But as industrial load becomes more and more preponderant as the controlling factor in the total load draft, it is quite conceivable that the shape of the total load curve, during the working day hours on the days of annual peak, may become such that the fitting in of a given amount of hydro energy will not show quite as large a peak ordinate as Mr. Funk obtains in his example.

The chief interest of the public today in hydro questions concerns the utilization of a number of outstanding large projects all of which are located some distance away from suitable load centers. Mr. Place's discussion serves as a timely reminder that in the search for substantial savings from large-size hydro projects we should not overlook some worth-while economies offered by the small hydro sites. Under the favorable conditions described by Mr. Place, the location value, i.e., local peak service, savings in transmission and transformer capacity, in voltage-regulating equipment, certain service protective features for the locality, saving of energy losses, etc., may add an appreciable sum to the pure steam replacement value, figured for the energy output from small hydro plants located in the vicinity of small load centers. Certain practices developed in recent years are steps in the right direction favoring development of such small plants having location value; this refers to automatic or semi-automatic operation, to the omission of the costly governors, to the curtailing of structural investment by the use of outdoor equipment, etc.

Mr. Eberhardt's fear that the speed variation of 1 per cent (± 0.5 per cent) in connection with the load signaling by frequency may be objectionable for some industrial drives is not warranted, judging from six years' experience on the Baltimore-Holtwood system. During that period continuous service was given to a large number of industries that are notoriously sensitive to speed changes, such as cotton spinning and weaving mills, silk mills, watch factories, paper mills, etc. What these mills strongly object to are sudden changes in frequency, even smaller than half of one per cent; or, in other words, within certain reasonable limits it is the rate of speed change and not the gradual variation of motor speed that is causing the trouble in the manufacturing processes. For example, the average-type induction motor will drop about one per cent in speed on a 10 per cent voltage drop. Practically every voltage disturbance will therefore cause a sudden speed change. Or the method quoted by Mr. Place for the automatic starting of hydro units on a drop in frequency, will probably cause a relatively sudden change in speed, particularly at times of sharply ascending load drafts. But the change in speed caused by the frequency signaling scheme is very gradual as can be noted from the frequency chart shown in Fig. 6, where the total one per cent change in frequency is distributed gradually over a period of forty minutes. This probably is the explanation why the frequency signaling system has not caused any complaint from the above-mentioned industrial drives sensitive to speed changes.

With certain limitations the frequency signaling scheme as used on the Holtwood-Baltimore load system for controlling the division of load between hydro and steam plants, can also be adapted to an operating system comprising several generating stations. It should only be necessary to set the governors in all the hydro plants for the half of one per cent higher range of frequency and all the steam governors for the lower range. The division of load among several run-of-river hydro plants is of no consequence during excess flow, but the distribution of generation between the steam plants must be adjusted either by approximately prearranged schedule or guided by load-dispatching communications.

The scheme becomes less suitable for a hydro-steam system where the hydro source comprises not only run-of-river plants but also peak stations. As pointed out by Mr. Eberhardt, this signalling plan by frequency can probably not be used to advantage where a number of load systems, each fed by a multiplicity of hydro and steam generating stations, are tied together loosely by long transmission lines. It is also not suitable for those systems where the electric distributing company endeavors to sell time on the basis of accurately maintained average frequency.

Messrs. Tefft and Eberhardt point out the desirability of letting the steam units carry the momentary fluctuations of load even during low flow when the hydro plants operate on the peak por-

tion of the system load. As explained in the paper, this is more or less automatically accomplished by the inherent difference in speed of governing of hydro and steam units. They also suggest that a hydro unit, whenever possible, operate at or near its best efficiency gate opening. This is a good operating rule, well supported by theory as well as by practical results. On large systems the effect on steam economy of this mode of operation will usually be found quite negligible.

Mr. Powell emphasizes the importance of developing a routine method for actually carrying out the ideal of hydro-steam coödination in every-day operation, which should aim at a minimum overall expenditure for energy, for the combined sources of power. This can be facilitated by first expressing separately the efficiencies of the various primary energy supplies in terms of a common unit.

For the steam plant the most suitable unit is "Lb. of Coal"; for the hydro plant it is not the amount of water but the amount of energy in kilowatt-hours metered at the load center that is obtained every day per unit of water. On the Baltimore-Holtwood load system this latter term has been known under the name of "Delivered Hydro Equivalent." The extent to which the different methods of load distribution affect the efficiencies and units of supply at the various stations must be well understood and their cross-relations should be expressed in simple curves or equations suitable for every-day use of the operating force.

No. 1966

MECHANICAL FEATURES AFFECTING THE RELIABLE AND ECONOMICAL OPERATION OF HYDROELECTRIC PLANTS

By E. A. Dow,[1] Worcester, Mass.
Non-Member

This paper deals with the subjects of flashboards, trashracks, rack rakes, sluice gates, headgates, penstock valves, air vents, and discharge-measuring devices, from the standpoint of economy and reliability of operation, based almost entirely on the New England Power Company's practice. It stresses the importance of reducing head losses and leakage and adhering to strength and simplicity in the design of hydraulic equipment.

AN EXHAUSTIVE treatise on mechanical features affecting the reliable and economical operation of hydroelectric plants would be entirely too long for presentation at a Society meeting, and this paper makes no attempt to cover the ground completely. It merely presents a series of comments and opinions on some features directly related to the subject. Practice varies rather widely with localities and conditions and there will doubtless be disagreements with many of the statements made. The following pages contain little of an original nature, but it is hoped that they will serve to call forth in discussion the ideas of other engineers engaged in similar lines of work.

2 By far the greater part of the author's experience has been with the plants of the New England Power Company, and the practices herein set forth have developed during the evolution of the New England Power System. A brief description of the system will be helpful to an understanding of the matters discussed.

The New England Power System

3 As will be evident from Fig. 1, which shows a map of the system, the New England Power Company serves a highly

[1] Mechanical Engineer, New England Power Construction Co.

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developed industrial territory, comprising central Massachusetts, southern Vermont and New Hampshire, and northern Rhode Island. It owns seven hydro plants, of an aggregate capacity of 115,000 kw., as well as a small steam station, and has several important interconnections with other systems, both steam and hydro. The total length of high-tension transmission lines is about 600 miles. In 1924 the system output was 563,000,000 kw-hr. and the maximum peak load was slightly over 150,000 kw.

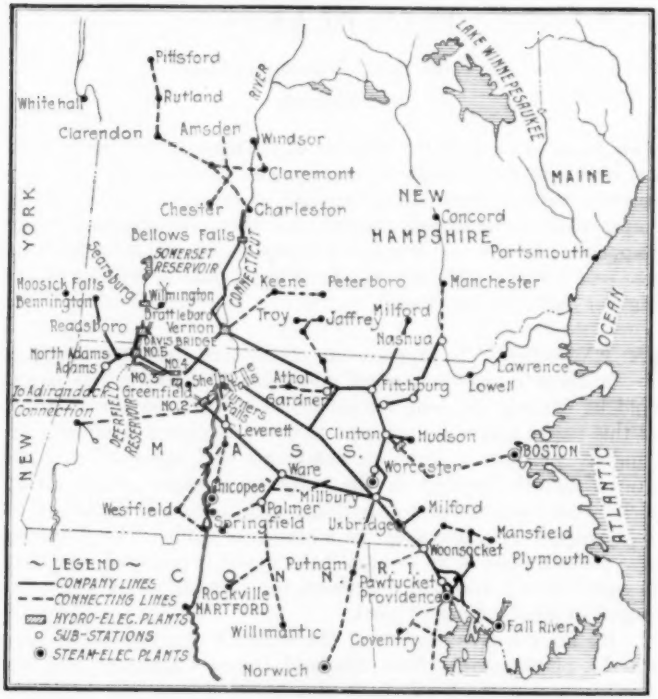

FIG. 1 THE NEW ENGLAND POWER SYSTEM

4 Power was first produced by the company in 1909 when the Vernon Station went into service a few miles below Brattleboro on the Connecticut River. The original installation at this plant consisted of eight 2500-kw. vertical three-runner units under 34-ft. head. Two modern single-runner 5000-kw. units were added in 1919, making the present station capacity 30,000-kw. The Vernon plant was followed in 1912 by three plants on the Deerfield River at Shelburne Falls, Mass., each having three double-runner horizontal units of 2000 kw. capacity. The heads on these plants are 60, 66, and 64 ft. The following year, 1913,

saw the completion of the Somerset Reservoir at the headwaters of the Deerfield, a storage proposition consisting of a 110-ft. earth dam impounding 2,500,000,000 cu. ft. of water. No power is developed at this dam, but the stored water is used through all the Deerfield plants. The next plant to be completed is known as No. 5, about three miles above Hoosac Tunnel, which went into service early in 1915. The equipment here consists of three single-runner horizontal units set in cast-iron scrolls and operating under 240 ft. head. The station capacity is 17,000 kw. Passing over a small temporary plant at Readsboro, the next plant on the Deerfield was completed in 1922 at Searsburg. This is a single-unit

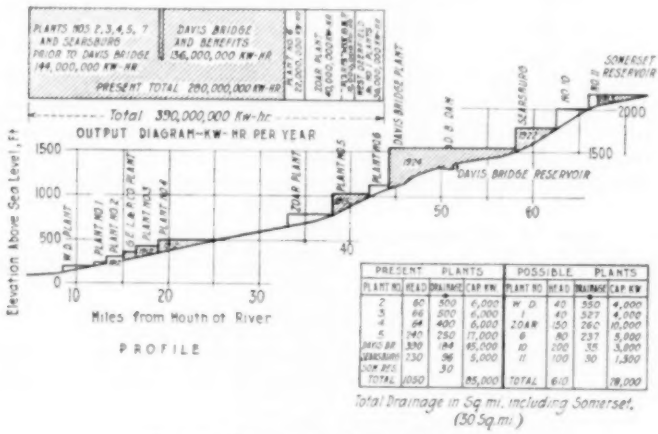
PLANTS NOS 2, 3, 4, 5, 7 AND SEARSBURG PRIOR TO DAVIS BRIDGE 144,000,000 KW-HR
DAVIS BRIDGE AND BENEFITS 256,000,000 KW-HR
PRESENT TOTAL 280,000,000 KW-HR
Total 390,000,000 Kw-hr
OUTPUT DIAGRAM-KW-HR PER YEAR
Elevation Above Sea Level, Ft
Miles from Mouth of River
PROFILE
DAVIS BRIDGE RESERVOIR
SOMERSET RESERVOIR
PRESENT PLANTS
POSSIBLE PLANTS
Total Drainage in Sq mi, including Somerset. (30 Sq mi.)

FIG. 2 PROFILE OF DEERFIELD RIVER

installation having a capacity of 5000 kw. under 230 ft. head. It is of interest by reason of being a completely automatic plant which has operated with entire success in spite of hydraulic complications such as 18,000 ft. of wood-stave conduit, differential surge tank, etc. The latest addition to the Deerfield list is the Davis Bridge Plant near Readsboro, Vt. This is primarily a storage proposition and has an earth dam 200 ft. high forming a reservoir of 5,000,000,000 cu. ft. capacity, but it is also the largest power plant on the system. The original installation consisted of two 15,000-kw. units operating under a head which varies between 300 and 390 ft. A third unit of identical design is now being installed. The turbines in this plant are of the single-runner vertical type, set in plate-steel spiral casings. A very complete description of this development is given by Mr. A. C. Eaton[1] in

[1] Hydraulic Engineer, New England Power Construction Co.

the *Journal* of the Boston Society of Civil Engineers, January, 1925.

5 Fig. 2 shows a profile of the Deerfield River, on which present and possible developments are indicated. As will be seen from the tabulation in the lower right-hand corner, the total head at present developed is 1050 ft. The watershed is very completely controlled and the flow is equalized to an unusual extent. For a large percentage of the time every cubic foot of water is of value, which means that high efficiency in apparatus and minimum head losses in racks, gates, etc., are of the greatest importance. The Deerfield plants are operated as far as possible on a 9-hour schedule and are shut down completely for a number of hours at night. Under these conditions it becomes necessary to reduce leakage during shutdown periods to the lowest possible point and a great deal of attention has been devoted to this matter.

6 Perhaps the most logical way to handle the discussion of a series of more or less disconnected items is to commence with those features pertaining to dam or intake and follow along downstream with the water. This introduces us to one of the most difficult and perplexing subjects first, namely, spillway-crest control.

Flashboards, Crest Gates, etc.

7 The ideal crest-control device would have the following characteristics:

1 It would be entirely automatic in operation
2 It would hold the water elevation above the dam between definite and comparatively narrow upper and lower limits
3 It would be tight
4 It would be unaffected by ice or other interferences with operation
5 It would leave an unobstructed crest under flood conditions or at least present openings of sufficient size to obviate danger of clogging
6 It would be reasonable in cost.

8 The devices actually in general use include simple flashboards supported by pipes or pins designed to bend over and clear the crest when the water surface reaches a predetermined elevation, rather elaborate arrangements balanced hydraulically or by counterweights and dropping or rising automatically with a change in water level, and crest gates of the sector or sliding type which depend entirely on the human element for their operation. Probably no available type combines all the desirable features, and for each installation a compromise choice must be made and that arrangement adopted which most nearly satisfies the conditions at hand. This is not the place to discuss the various possibilities in detail, particularly since the subject has already

been covered fairly thoroughly in the 1924 report of the Hydraulic Power Committee of the National Electric Light Association.[1]

9 With the exception of those for the storage developments, all the spillways of the New England Power System are equipped with the simplest type of flashboards, ranging in height from 4 to 8 ft., and in one case even 10 ft. The boards are supported by steel pipes of various sizes, weights, and grades, selected to suit individual conditions, and it has been found that failure takes place with very satisfactory consistency at the desired pond elevations. When there is sufficient warning of an impending flood, boards are removed, otherwise they are allowed to go and are lost, which occurs on an average of probably once a year. Removal and replacement of boards is effected by means of boats, assisted in some cases by cableways. All plants are provided with sluice-gate capacity properly proportioned to the drainage area, so that shortly after the peak of a flood has passed the pond can, by means of sluice gates and turbines, be sufficiently lowered to permit of replacing boards. Satisfactory tightness is maintained by calking with cinders, which may seem somewhat surprising, but the flashboards are assembled in sections, which aids in this respect. The system is by no means perfect but all things considered is preferable to any other which has as yet been devised. It is low in first cost, probably no more expensive as regards maintenance and fixed charges than the more elaborate arrangements, and more reliable as a protection against flood damage than any device depending entirely on human vigilance. There are certain dangers to be guarded against, however, among which are a temptation to brace pins to carry higher boards during months when the chance of sudden floods seems remote, and the tendency toward the formation of ice on the downstream face of boards and the upper part of the dam crest. If this is allowed to accumulate in heavy masses it may reinforce the pins so that failure will not occur until the pond level rises materially above the desired elevation.

10 At Somerset and Davis Bridge conditions affecting flashboards are materially different from those at other plants. The large storage capacities in the upper levels of the reservoirs eliminate the possibility of a dangerously sudden rise in water elevation, and on account of the great value of the stored water it is out of the question to draw down materially to replace flashboards after a flood. A control is required which can be replaced with a considerable depth of flow over the crest, yet it must be of such a type that it can be removed with certainty, by one man if necessary, in spite of ice, etc. Because of the lack of velocity of approach to the spillway of a large reservoir there is much

[1] Publication No. 24-28.

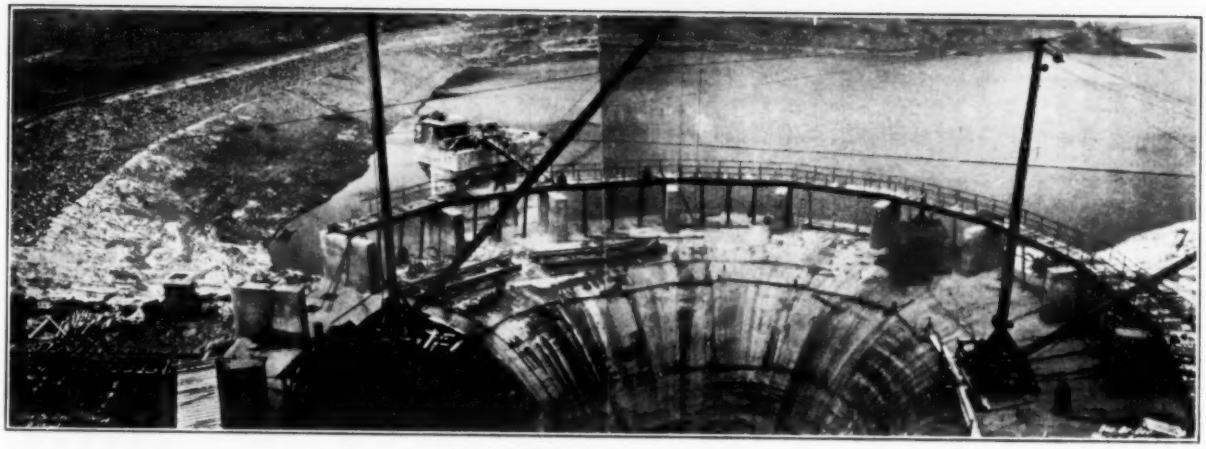

FIG. 3 DAVIS BRIDGE SPILLWAY

less danger of spillway openings becoming clogged with floating trees, ice floes, etc., than in the case of a river spillway, consequently the objection to piers on the crest is greatly reduced. The type of control which seemed best for both Somerset and Davis Bridge consists of stoplogs supported by needle beams which can be tripped from a bridge spanning the openings between spillway piers. Fig. 3 shows a general view of the arrangement at Davis Bridge, the details of the device being shown by Fig. 4. The spillway is circular in plan, 160 ft. in diameter, the discharge passing down a shaft 22½ ft. in diameter and out through a horizontal tunnel located about 170 ft. below the crest. There are 16 radial piers and the span between each pier is broken up by three needle

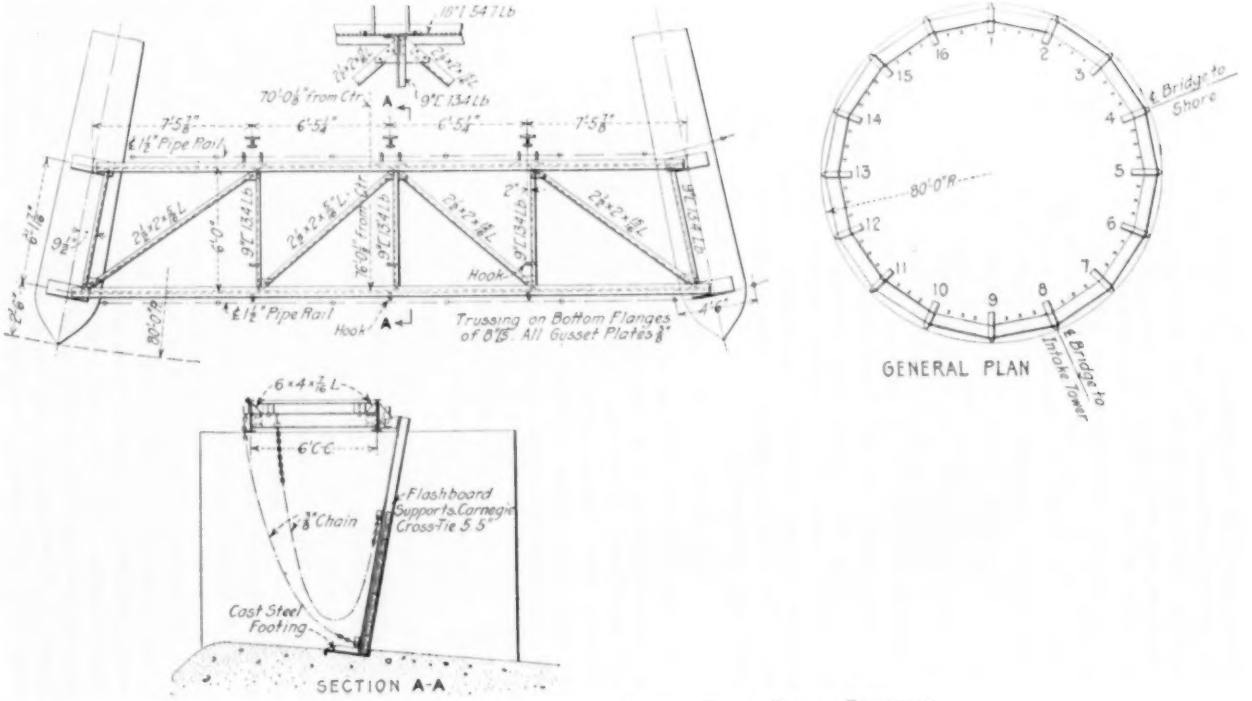

FIG. 4 DETAILS OF FLASHBOARD DEVICE, DAVIS BRIDGE SPILLWAY

beams supporting the stoplogs. The beams can be tripped from the bridge by means of the latch at the top, thus releasing the stoplogs. They are still held by the chains, however, and can be pulled up and replaced at any time, the tops being returned to the latch and the bottom ends allowed to drop into the sockets at the bottom. Conditions are such that it will hardly ever be necessary to trip the needles and lose the stop planks at either Davis Bridge or Somerset. Ordinarily boards could be pulled up from the bridge to accommodate the flow.

11 Many other devices were considered for the Davis Bridge spillway, among which were radial gates and ordinary pin-supported flashboards. The latter involved the use of a bulkhead of curved form which could be moved around the spillway on a circular track and used to shut off the openings between piers, one after another, so that flashboards could be replaced in one bay at a time with water flowing through the remaining openings. The Tainter gates were discarded because of the cost and the possibility of their becoming inoperative from freezing at a critical time. The other arrangement presented too many difficulties in design and construction.

Sluice Gates and Accessories

12 The value of sluice gates of some type suitably proportioned as regards capacity to the drainage area of the plant has already been mentioned. A few remarks about types and design may not be out of place. Either surface or submerged sluices can be used and both have certain advantages, but for handling fairly large flows the submerged type ordinarily proves more economical and generally attractive than the other. Discharge may take place directly into the air or below tailwater. The first arrangement permits of convenient inspection for leakage, etc., while the latter is less subject to trouble from freezing. The commonest kind of submerged sluice consists simply of a cast-iron frame and sliding gate provided with a screw stem and mechanical hoist, or with a pressure cylinder. The latter is very much to be preferred when the operating mechanism is located in a damp place (say, a gallery in a dam) as a mechanical hoist, particularly if electrically operated, is extremely difficult to keep in proper maintenance under such conditions.

13 A typical installation of this kind is illustrated in Fig. 5. This shows one of the seven 9- by 7-ft. Vernon sluice gates which have an instructive history. The original gate installation at Vernon consisted of eight gates of a manufacturer's standard design, controlled by bronze stems and electrically operated mechanical hoists. The gates were rather light, the tensile stress in the cast iron running as high as 4000 to 4500 lb. per sq. in. Numerous difficulties were experienced with these gates for several years. The hoist capacity had been figured on too low a friction

coefficient and the hoists were overworked. The stems were correspondingly light and had probably been designed on the assumption that since they were made of bronze having the same tensile strength as steel they would be equal to steel as columns,

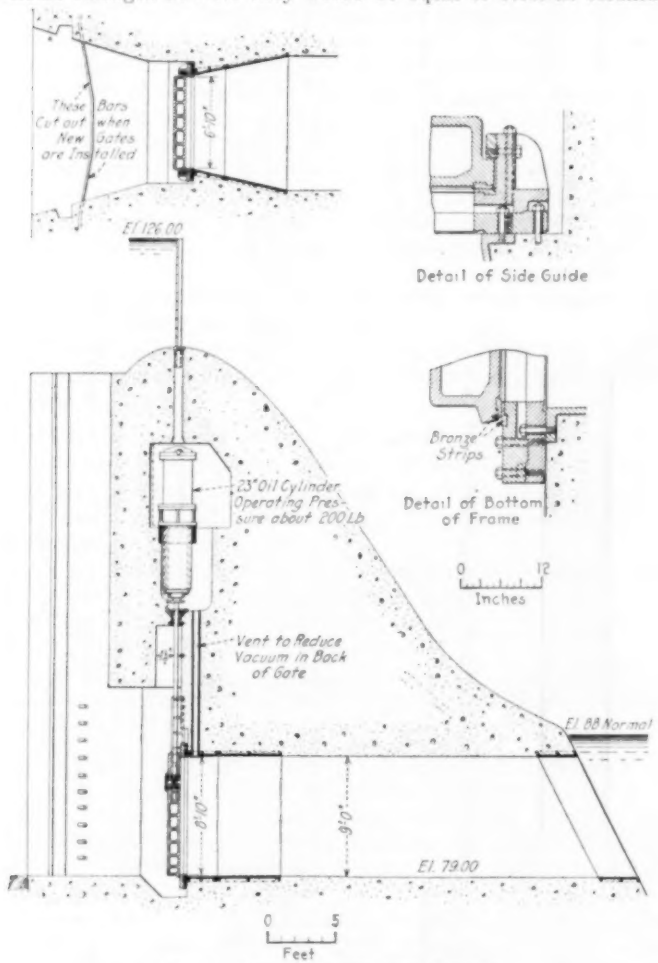

FIG. 5 VERNON SUBMERGED SLUICES

no attention being paid to the effect of a much lower modulus of elasticity. Several of the stems buckled. The gates and frames had bronze wearing strips held in place by screws. Sections of these strips broke out, causing excessive leakage. The gates themselves gradually sagged and warped under the load and one of

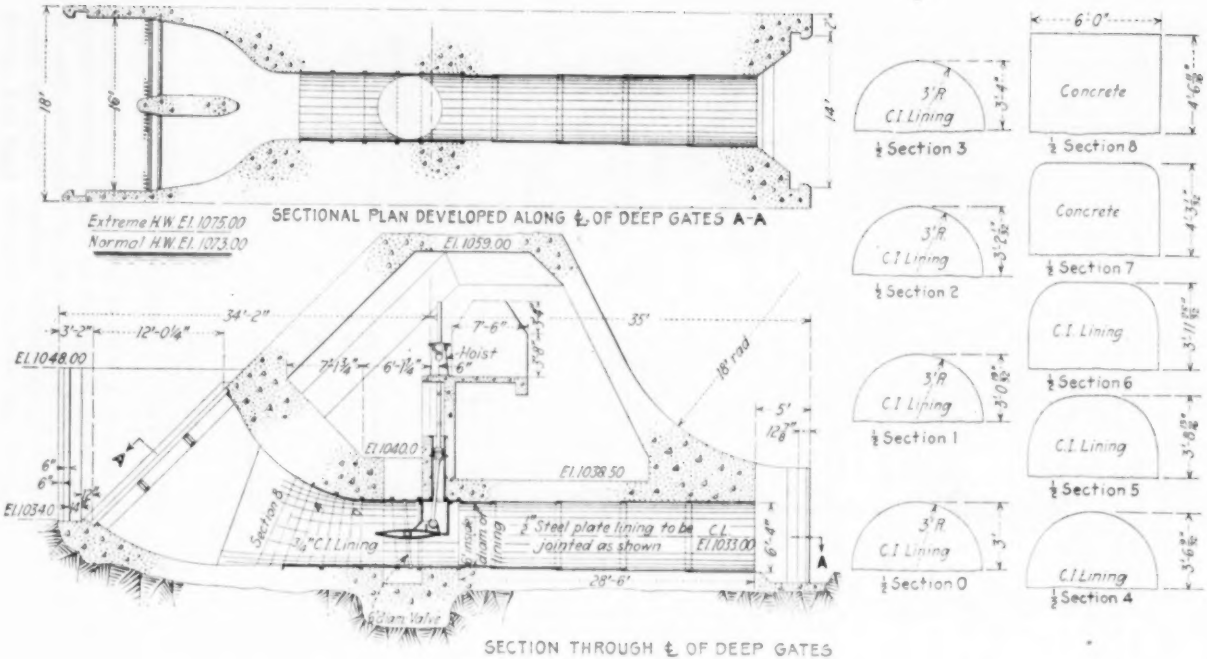

FIG. 6 BRASSUA LAKE VALVES

them finally failed and went out altogether, leaving the sluice wide open. The entire set of gates and frames were finally replaced with much heavier equipment especially designed to meet the Vernon conditions, using tensile stresses under 1500 lb. per sq. in. in cast iron and substituting oil-pressure operating cylinders for the mechanical hoists. Upon unwatering the discharge passages it was found that in several cases the cast-iron lining to which the gate frames were bolted had been loosened, cracked, and warped, probably from the effect of headwater pressure between the lining and the concrete, combined with vibration. In connection with the replacement another source of trouble was eliminated by the removal of a heavy coarse grillage of horizontal bars originally installed to prevent stoppage of the openings by water-logged stumps, etc. It had been found that this grillage did more harm than good by catching trees and logs after they had passed part way through and holding them where they would interfere most with the closure of the gates.

14 From the Vernon experience several things were learned of general application in connection with sluice-gate installations. Among these are:

a That it does not pay to economize on first cost by using light sluice-gate equipment. The cost of replacement is excessive.

b That designs should be based on a coefficient of sliding friction of at least 0.75. This value has been encountered in actual experience.

c That bronze should not be used in compression when it can be avoided. Steel protected by a bronze sleeve is preferable.

d That metal liners below free discharge gates should be properly anchored back into the concrete.

e That if a grillage is used before a sluice gate it should be located sufficiently far upstream so that debris caught in it will not trail down stream far enough to obstruct the gate.

15 Plain sliding sluice gates can be expected, if properly designed, to operate satisfactorily for free discharge service up to heads of 50 to 75 ft., probably depending to some extent on size. For higher heads the vibration is likely to be excessive and some other type becomes preferable. Several recent installations have made use of a modified design of butterfly valve, known as the disk-arm pivot valve.[1] This valve, to which further reference will be made later, differs from the ordinary butterfly in that it is controlled by a rod connecting with an arm cast integral with the disk rather than by a crank on the end of the spindle. Such an arrangement makes possible an extremely strong and rigid operating mechanism with a consequent elimination of vibration under high velocities. The United Hudson Electric Company has

[1] Patents pending.

an installation consisting of six 72-in. valves of this type operating under 110 ft. head at their Sturgeon Pool plant. These valves are free from vibration at all openings. Fig. 6 illustrates a setting used at the Brassua Lake storage dam, where four 72-in. disk-arm valves are being installed under a head of only 37 ft.

16 A feature of the greatest importance in free-discharge sluice design for any considerable head is the water passage below the gate or valve. It is a common but decidedly questionable practice to flare this passage to increase the discharge by reducing the velocity head lost at exit. Under low heads a considerable increase in discharge capacity can be obtained by this expedient, but the percentage effect rapidly falls off as the head goes up. In the majority of cases a very trifling flare suffices to reduce the pressure at the valve to absolute zero and the most constricted part of the passage becomes the neck of the bottle. It is never possible to increase the discharge beyond that corresponding to the net head plus 34 ft. acting on this neck. Designs in which the pressures approach absolute zero result in a greatly increased tendency to vibration, and the combination of low pressure and high velocity is ideal for the pitting of metal and erosion of concrete. Vents admitting air to the discharge passages immediately below valves or gates serve to reduce the tendency to vibration and pitting somewhat, but are not a complete cure for the trouble. Increased maintenance expense due to a flaring passage is likely to eat up quickly any saving in first cost made by the use of a smaller gate or valve. The author knows at least one prominent engineer who advocates actually reducing the discharge passage below the control apparatus in a sluice in order to produce a nozzle effect and avoid erosion in the passage itself. For the higher heads it is beyond question that the control valve should, whenever possible, be located at the downstream end of the water passage and be allowed to discharge freely into the air.

Racks and Accessories

17 In most installations the water leaving the pond or reservoir passes through racks whose function is to screen out debris which might enter the turbines and clog or damage them. A satisfactory rack must stop all trash of objectionable size but must pass the required amount of water without excessive head loss.

18 This seems like a simple specification to meet, but experience shows otherwise, and clogged racks have probably been responsible for more serious hydro-plant shutdowns than any other one factor. The design of a suitable rack involves the consideration of the size and type of turbine to be protected, the character of trash to be taken care of, the means to be employed for cleaning, allowable head loss, ice conditions, and any special local peculiarities. In general the rack area should be as generous

and the velocities as low as possible, as difficulties of all kinds increase rapidly with higher velocities. The rack velocities at the New England Power plants range from well under one foot per second at Davis Bridge to 3.6 ft. per sec. at some of the Shelburne Falls intakes. In general satisfactory results may be expected from velocities in the neighborhood of 1½ ft. per sec. figured on the gross rack area. At average reservoir elevations and loads the

FIG. 7 VERNON RACK RAKES

rack velocities at Davis Bridge are extraordinarily low, the racks at this intake having been carried to the top of the intake tower so that it would be possible to rake at least a certain minimum area of them with water at the highest elevation. Due to drawing from a very large quiescent body of water very little trash reaches these racks and experience to date indicates that seasonal cleaning only will be necessary at this plant.

19 Under New England conditions the two things most likely to interfere most seriously with keeping racks in operation are

a run of frazil ice and a run of leaves. The former is most to be feared and is apt, under certain conditions, completely to clog racks in spite of all measures taken against it. This is true even with rather widely spaced rack bars, and when the worst comes to the worst about all that can be done is to pull racks and take a chance on trash which may enter the turbines. Ordinarily it is possible to cope with leaves by continuous raking and if bars are spaced about 2½ in. or more apart most of the leaves pass through and cause little trouble. This spacing, however, is greater than is safe with many wheels.

20 An interesting subject which has been receiving some attention during the last two or three years is the matter of mechanical rack-cleaning devices. It is possible to rake racks satisfactorily by hand to a depth of 15 to 20 ft., but with depths of 30 to 40 ft., such as are being found necessary in large low-head plants, hand raking is out of the question. Several mechanical rakes are now available and Fig. 7 shows a design which has been worked up in coöperation with the manufacturer of one of these rakes for use at the Vernon Station. This design involves a traveling motor-operated rake which can be moved to any desired bay and dropped down in suitable fixed guides with which the traveler is lined up. The rake is approximately 20 ft. wide and cleans an entire bay at one operation. During the spring high water the rack loss frequently amounted to several feet and under the worst conditions it was necessary to shut a unit down in order to rake it successfully, which meant that one unit was out of service almost continuously. The annual loss in output at Vernon due to clogged racks has been estimated at about 3,000,000 kw-hr. and the cost of hand raking has amounted to from $2500 to $3000 per year. It is evident that the total annual loss is sufficient to justify a considerable expenditure for a device able to eliminate any appreciable percentage of it.

21 The matter of loading for rack design is something on which practice appears to vary widely. The difference of head to be expected due to clogging of course varies with local conditions, but for most of the New England Power plants it has been thought advisable to design the bars for the total head, using stresses up to the elastic limit. Supports are designed with a factor of safety slightly greater than that assumed for the bars, with the expectation that they may be used to support bulkheads for unwatering purposes. In the case of very deep racks unlikely to clog completely, such as those at Davis Bridge, a head difference of 15 ft. has been used.

Headgates and Operating Mechanism

22 Next in order for consideration are gates used at intakes to water passages, such as canals, tunnels, penstocks, etc., and com-

monly called "headgates." These are not ordinarily used for throttling or regulating purposes and the principles involved in their design and selection are not the same as those affecting sluice gates. Headgates are in general protective devices used to shut off the water passages and turbines in case of accident or when drainage is necessary for inspection, repairs, etc., and so far as possible, they should fulfill the following requirements:

1 They should be able to close in an emergency in spite of the maximum discharge which could occur under any probable condition
2 They should be reliable and reasonably rapid in operation
3 They should be sufficiently tight or so designed as to be easily calked so that leakage would not interfere with work to be done below them
4 They should be free from ice trouble
5 They should be safe against accidental opening
6 They should be foolproof and either protected from obstructions or designed to close against them without damage
7 They should cause a negligible head loss
8 They should be reasonable in first cost and maintenance.

23 In headgates as in other things perfection is difficult to attain, and most designs end up as compromises representing the best that can be done under limiting conditions.

24 Headgates naturally fall into two distinct classes depending on whether they operate under high or low pressure, and while they perform the same functions they are likely to vary widely in type. A fairly typical low-head installation is illustrated in Figs. 8 and 9, which show the gates used for the Vernon extension units put into service in 1921. Each of these units has a capacity of 5000 kw. under 34 ft. head and discharges about 1800 sec-ft. Each unit has two gates 17 ft. 6 in. high by 15 ft. 9 in. wide, operated by 28-in. oil cylinders.

25 In line with New England Power policy these gates are designed to close in an emergency under full head with no back pressure, so that it is possible to shut off the flow with the turbine gates open or with the turbine entirely wrecked. To aid in closing under emergency conditions the gates were made very heavy (120,000 lb. apiece) the construction consisting of a structural-steel frame filled with concrete. No skin plate was used. The weight of the gates plus the thrust of the hoists is sufficient to insure closure with a coefficient of friction of gate on guides of well over 0.75. They are opened with pressures equalized by means of filler gates.

26 These gates are of the plain sliding type and have seats of structural steel bearing on cast-iron frames. In common with most gates of this design they are not exceedingly tight but the

setting is such that except along the bottom edge they are easily calked by the use of cinders. They are sufficiently tight to satisfy all requirements in this particular installation.

27 Cylinder hoists such as are used for these gates have their own inherent advantages and disadvantages. They are extremely

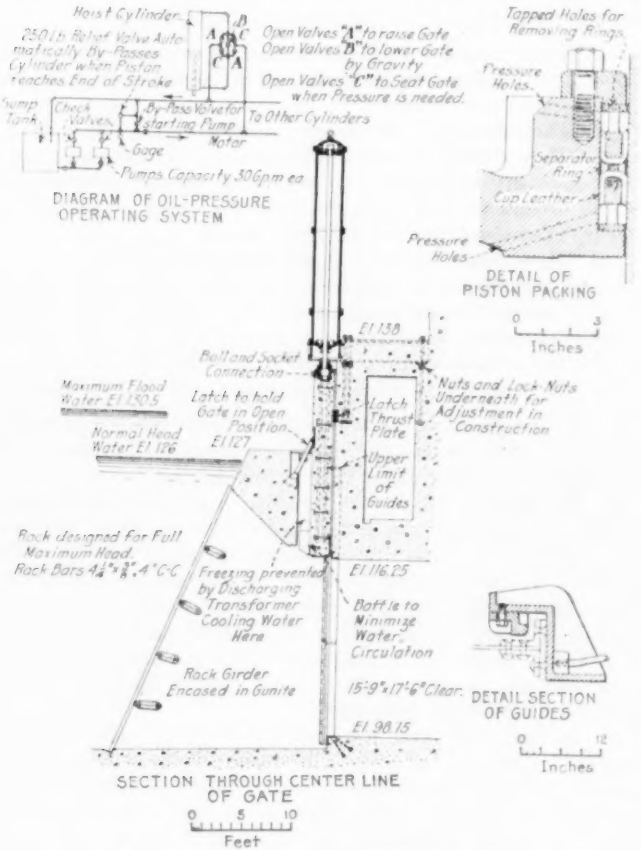
Hoist Cylinder
250 Lb. Relief Valve Automatically By-Passes Cylinder when Piston reaches End of Stroke
Open Valves "A" to raise Gate
Open Valves "B" to lower Gate by Gravity
Open Valves "C" to Seat Gate when Pressure is needed.
Sump Tank
Check Valves
By-Pass Valve for starting Pump
To Other Cylinders
Motor
Gage
Pumps Capacity 306 gpm ea
DIAGRAM OF OIL-PRESSURE OPERATING SYSTEM
Tapped Holes for Removing Rings
Pressure Holes
Separator Ring
Cup Leather
Pressure Holes
DETAIL OF PISTON PACKING
0 3
Inches
El 138
Bolt and Socket Connection
Maximum Flood Water El 130.5
Latch to hold Gate in Open Position
Normal Head Water El 126
El 127
Latch Thrust Plate
Upper Limit of Guides
Nuts and Lock Nuts Underneath for Adjustment in Construction
Rock designed for Full Maximum Head
Rock Bars 4 1/4" x 3/8", 4" C-C
Freezing prevented by Discharging Transformer Cooling Water Here
El 116.25
Bottle to Minimize Water Circulation
Rock Girder Encased in Gunite
15'-9" x 17'-6" Clear
El 98.15
DETAIL SECTION OF GUIDES
0 12
Inches
SECTION THROUGH CENTER LINE OF GATE
0 5 10
Feet

FIG. 8 VERNON EXTENSION HEADGATES

rugged and free from complication and are probably less subject to damage by careless or incompetent handling than any other type. A relief valve prevents the development of excessive pressure and nothing is injured if the gate closes against an obstruction, as the pressure simply builds up until the valve discharges. Under similar conditions a mechanical hoist would ordinarily buckle the stem, strip gear teeth, or otherwise wreck itself. As an offset

to these advantages, when used with heavy gates they involve some type of latch to sustain the weight when the gates are raised, and this is a complication if, as sometimes happens, it is desirable to provide for closure by remote control in an emergency. The Vernon gates are supported when open by struts of heavy pipe and it is necessary to raise them and release these latches before closure. If the cylinders are used in a cold climate in an exposed position they must be operated with some liquid which will neither freeze and damage the apparatus nor become so viscous as to refuse to flow. The Vernon cylinders are filled with a low cold-test hydraulic-jack oil. Stems must be smooth, otherwise the wear on

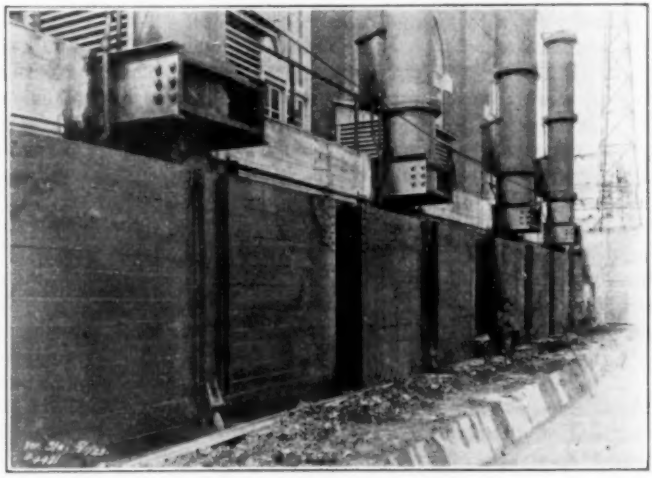

FIG. 9 VERNON EXTENSION HEADGATES

the stem packing and consequent loss of oil will be excessive. Cylinder walls must be smooth to prevent undue wear of piston packing. Ample pump capacity is necessary so that cylinders can be operated in spite of minor leakage past the piston. The New England Power Company has adopted a minimum pump capacity of 30 gal. per min. for such installations for this reason and greater capacity is very desirable, even though the desired speed of operation could theoretically be obtained with much less.

28 The handling of heavy as opposed to light gates by hydraulic cylinders introduces difficulties which most of us only come to appreciate through sad experience. The lowering of gates of this type is ordinarily accomplished by bypassing the operating fluid from the bottom to the top of the cylinder, the weight of the gate pulling on the piston developing the pressure which forces the fluid from the bottom to the top. This means that motion is

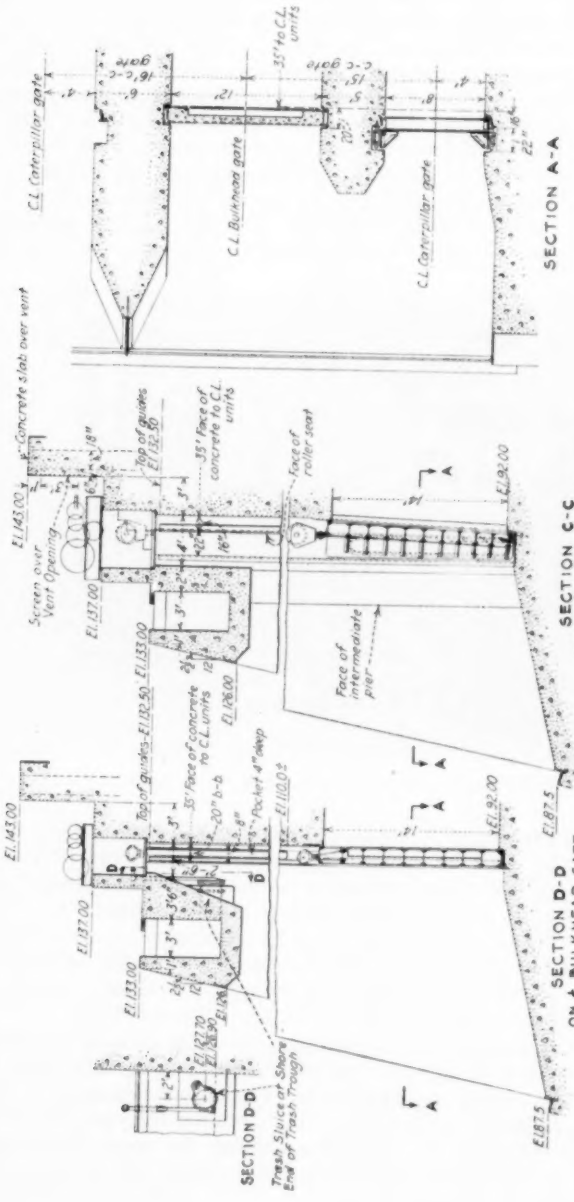

FIG. 10 INTAKE GATES, FARMINGTON RIVER POWER COMPANY

taking place *against* the pressure, and unless the piston leathers are provided with a separator ring of exactly the correct shape there is a tendency for the lower cup or U-leather to be drawn upward past the upper leather. The result is a severe folding and wedging against the cylinder walls, and the leather soon wears through. The author has a vivid recollection of struggling in the wee small hours with a headgate of his design which controlled a 24,000-hp. plant scheduled to go on the line at 7: 00 a.m., and which refused to open because the piston leather had cut through. The gate was only persuaded to operate by feeding the pump enough bran partially to calk the cut and check the leakage. A further peculiarity becomes evident if pressure is admitted to the top of a cylinder while a heavy gate is being lowered or is being sustained by the pressure below the piston. Under this condition the pressure from the pump is superimposed on that developed by the weight of the gate and the result is quite likely to be a pressure approaching twice that which the designer anticipated. This possibility should be avoided by the proper arrangement of connections or by relief valves. The above may appear to be a serious indictment of hydraulic-cylinder operation, but it is less so than it sounds. Most of the troubles experienced with hydraulic cylinders are vexatious rather than disastrous and are quickly rectified, while a wrecked mechanical hoist is an entirely different matter. Even a doubling up of pressure is unlikely to damage more than the pressure gage, as most cylinders and piping systems have a sufficient factor of safety to stand it. A mechanical hoist equally foolproof would be extremely difficult and expensive to build.

29 The Vernon headgates probably meet the requirements of the situation more completely than most similar installations. Men can work behind them without interference from leakage (generous drains take care of what cannot be stopped off by calking) and they can feel confident that the gates will not open of their own accord. The first cost of the installation was not unreasonable and the maintenance and deterioration are practically nothing. Interference with closure due to freezing is entirely eliminated by piping the slightly warm discharge from the water-cooled transformers into the gate wells, which are planked over better to retain the heat. The area is sufficiently large so that the head loss is practically eliminated.

30 A rather interesting intake using the so-called " caterpillar " gate made under the Broome patent is illustrated in Fig. 10. This shows the arrangement used at the Rainbow plant of the Farmington River Power Company now under construction near Hartford, Conn. The Broome gate is built with a seat inclined to the guides in such a way that the water pressure on the upstream side of the gate has a component in the direction of closure, so that the gate is self-closing if the friction is sufficiently low. In the caterpillar design frictional resistance is largely eliminated by the use

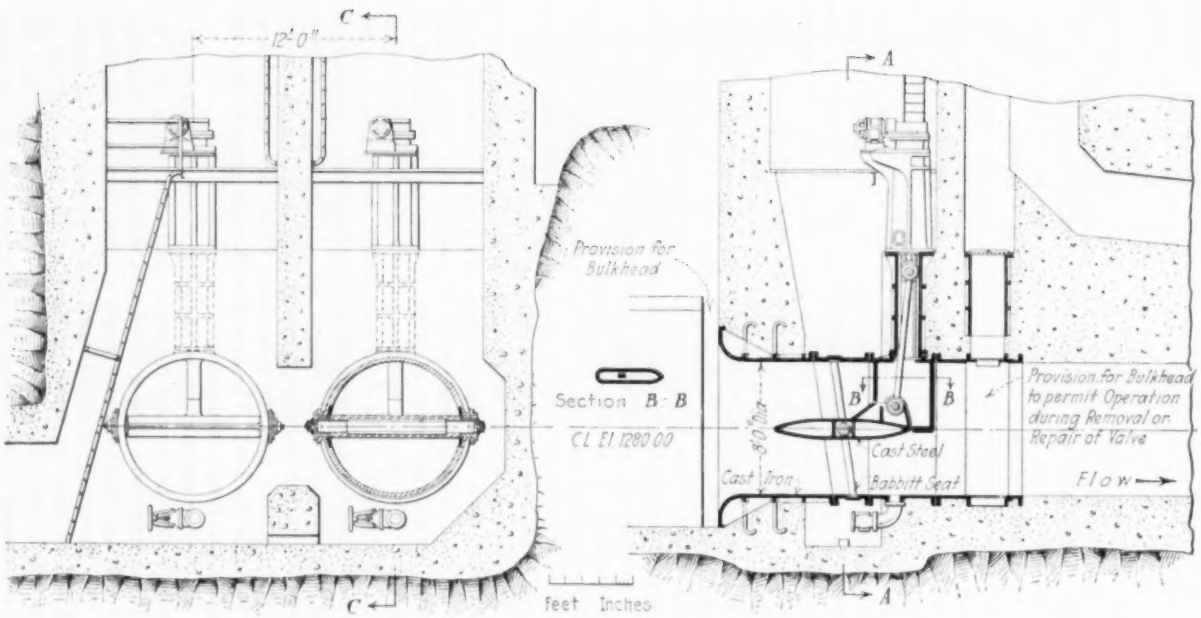

FIG. 11 DAVIS BRIDGE INTAKE

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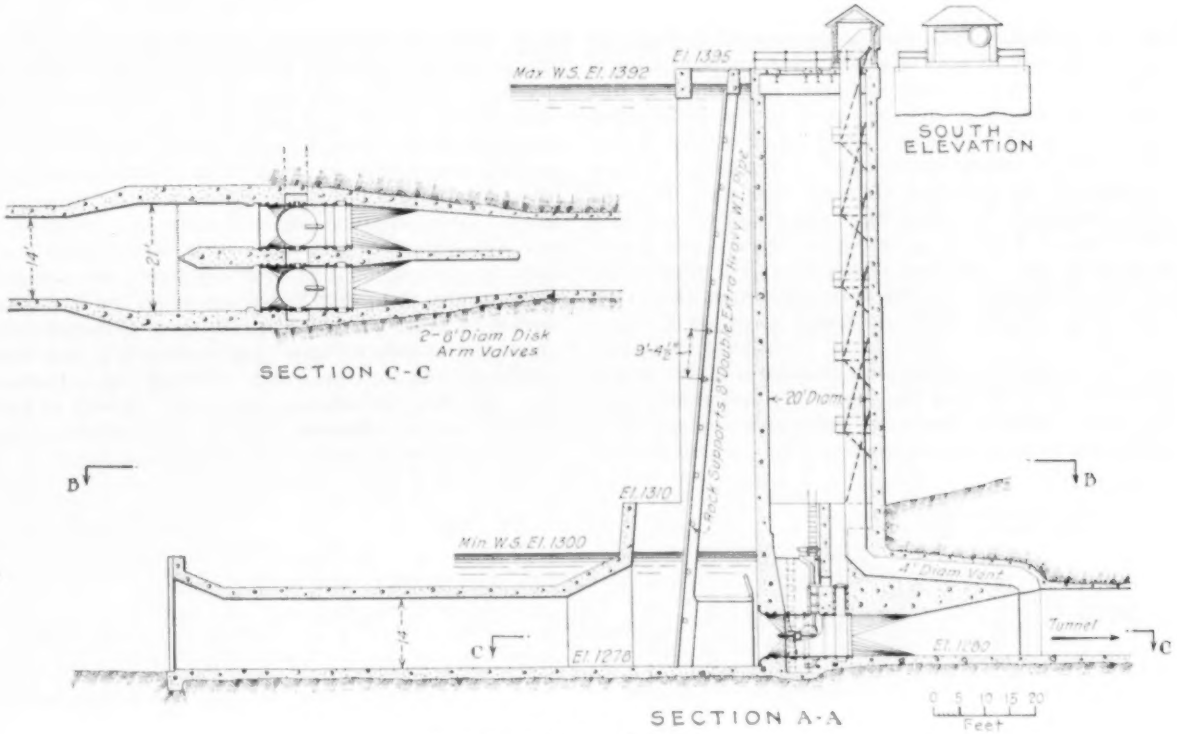

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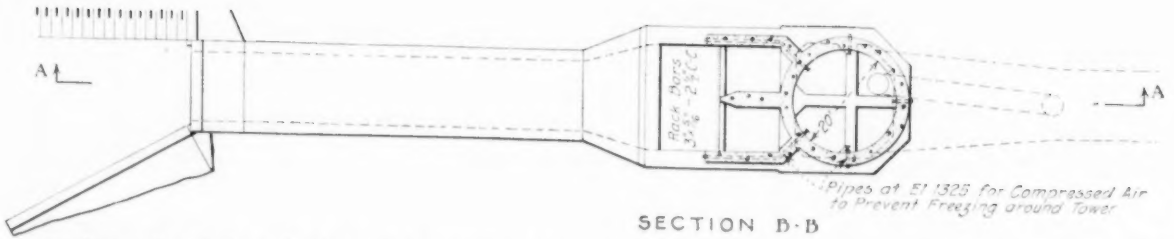


SECTION B-B

of an endless roller chain and a hoist of the tension type only is required, since it is not necessary to force the gate down in closing. The intake arrangement originally considered for the Rainbow plant called for two caterpillar gates each 14 ft. high by 10 ft. wide for each unit. Further study indicated, however, that the cost could be greatly reduced by the use of one sliding gate 14 ft. high by 12 ft. wide and one caterpillar gate 14 ft. high by 8 ft. wide. The combined area is the same and the sliding gate will be weighted with concrete so that it will be heavy enough to close by its own weight under full turbine discharge, with the caterpillar gate open. In other words, it will slide against the friction resulting from a head difference sufficient to put the entire flow through the smaller gate. When the sliding gate is down the caterpillar gate will be closed under full head and of course the latter will be opened first when it is desired to fill the scroll. This installation costs about two-thirds as much as the design using the two caterpillar gates and is equally good, the only requirement being that the sliding gate be operated first in case it is necessary to close the headgates with the turbine gates open.

31 A layout in which the gates operate under relatively high head is shown in Fig. 11, which illustrates the intake from the Davis Bridge reservoir. The water level in this reservoir fluctuates over 90 ft. and it was of course necessary to locate the gates below the minimum water elevation. Actually, they are subjected to a maximum head of about 110 ft. The intake design adopted consists of a dry concrete tower 117 ft. high having an inside diameter of 20 ft., in the bottom of which are located two 8-ft.-diameter valves. These valves are motor-operated and are of the disk-arm type previously referred to, which as a matter of fact was developed for this

particular location. They were adopted in preference to rectangular sliding gates which were less attractive because of the excessive hoist capacity required for reliable operation under the existing head, difficulties to be expected from damaged seats under the heavy pressures, etc. In valves of the butterfly type the danger of cutting seats is minimized, since no sliding takes place on them, and in the design adopted the seats are of babbitt, which is very resistant to erosion and can easily be repaired in place if damaged.

32 In connection with this installation considerable experimental work was done to determine the hydraulic characteristics of pivot valves and it was found that under any considerable velocity they have a powerful tendency to close as soon as the disk is moved from the wide-open position. The construction adopted was preferable to the standard butterfly because the use of an arm cast integral with the disk, permitting direct connection of the operating mechanism, consisting of stem, connecting rod, and crosshead, facilitated the development of a design insuring reliability of closure even under free spouting conditions, such as might occur with a blowout in the tunnel or other accident. In this installation the disk-arm type was attractive for the additional reason that the central hoist arrangement made possible a closer spacing and consequently a smaller tower than could have been used had the valves been controlled from the ends of the spindles.

33 These valves have proved extremely tight, probably fully as tight as sliding gates, and certainly much more so than the standard butterfly. On several different closures one of them was found to show a leakage too small to measure, while the other leaked between 5 and 6 gal. per min. under 100 ft. head. The latter is shown in Fig. 12, the leak being clearly seen as a small jet issuing from the upper right-hand quarter. This unusual tightness is attributable to the fact that the thrust of the hoist opposes the water pressure on the trailing half of the disk and prevents deflection, forcing the edge tightly against the seat around the whole circumference.

34 Davis Bridge is the largest single plant on the New England Power System, and continuity of service is of the greatest possible importance. For this reason every precaution has been taken to insure against interruption, and provision has been made for closing off either valve for repairs while the plant is supplied through the other. This is done by the use of upstream and downstream bulkheads which can be quickly installed when required. The valve housings are built with beveled flanges so that an entire housing and disk can be removed if necessary.

35 In order to economize on size of tower and cost of valves, fairly high velocities are used in the Davis Bridge intake design. At the maximum expected discharge of 1600 sec-ft. the velocity

through the gross area of the two 8-ft. valves will be 16 ft. per sec., the actual velocity at the most constricted section reaching about 19 ft. per sec. The water passages were carefully designed, however, particularly in the flaring portions, and the actual head losses are gratifyingly low. A test at a discharge of 1000 sec-ft. gave a loss from pond to the beginning of the standard tunnel section of only 0.75 ft., which indicates that under average loads the intake loss will amount to less than one-half of one per cent of the total head.

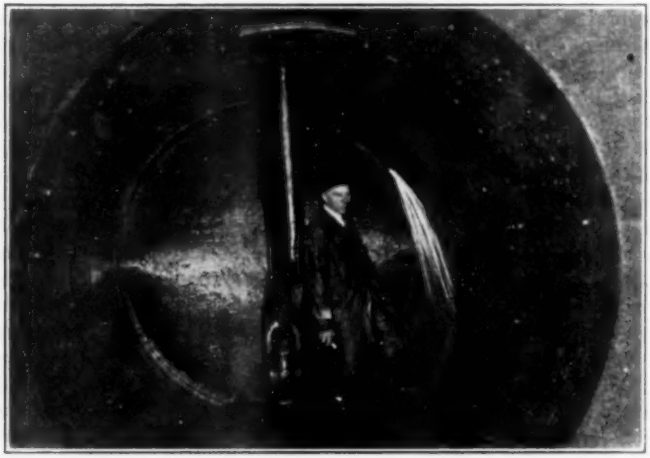


FIG. 12 96-IN. DISK-ARM VALVE, DAVIS BRIDGE INTAKE

PENSTOCK AND TURBINE-INLET VALVES

36 It frequently happens that water is supplied to a hydro station through a single tunnel, conduit, or canal, which divides into several penstocks to feed individual units. Under these conditions it becomes necessary to provide means for shutting off the water from each unit separately in order to avoid shutting down the entire plant whenever a single unit must be unwatered. This situation arises at two of the New England Power plants — at No. 5, where the water is carried through about three miles of conduits, canals, and tunnels to a simple surge tank, whence it passes into three steel penstocks, and at Davis Bridge, where the water is led from the reservoir to the point of subdivision through two and a half miles of 14-ft.-diameter tunnel. At the tunnel outlet a branch connects with a Johnson differential surge tank and immediately below this a plate-steel manifold distributes the water to three 9-ft.-diameter steel penstocks about 600 ft. long.

37 As usual under such conditions, the Davis Bridge situation brought up the question of whether to use valves at the top or bottom ends of the penstocks, or at both top and bottom. Valves at the top would permit unwatering any penstock for inspection, painting, or other maintenance work, without interfering with the operation of other units, and also would afford a very desirable safeguard by facilitating a quick shutting off of flow in case of an accident to a penstock. (The main intake gates could of course be closed in the event of a break, but the huge volume of water in the tunnel would have to drain out before the flow would stop.) On the other hand, valves at the top could not be used satisfactorily to reduce leakage during nightly shutdown periods, since this would involve too frequent draining of the penstocks which would probably be detrimental on account of increased corrosion. It would also be unsatisfactory as an operating proposition, for the time required to fill a penstock would render it impossible to put a unit on the line quickly in an emergency. The problem was studied carefully and the final decision was to use valves at both ends. This was in line with the policy adopted at Plant No. 5, where the same situation had been considered several years earlier.

38 Before passing to other matters it may be worth while to lay somewhat greater emphasis on the importance of reducing leakage in connection with such a plant as Davis Bridge. By actual measurement the leakage through the wheel gates of each unit when first installed and in perfect adjustment was found to be about 9 sec-ft. This is an exceptionally low value and experience indicates that it may be expected to increase considerably as time goes on. The Davis Bridge units are operated in general on a 9-hour basis and during the remaining time, at least 5000 hr. per year, they are shut down. Assuming a leakage of 15 sec-ft. per unit, or 45 sec-ft. for the plant, and an average head of 345 ft., the yearly loss of output would amount to something over 500,000 kw-hr. Since the Davis Bridge water is almost completely used, this is a real loss and the reduction of leakage by the use of valves at the turbine inlets is sufficient to justify the expenditure for such valves entirely aside from the additional considerations of time saved in getting into and out of a turbine when necessary and various miscellaneous conveniences. The same conditions occur at the Searsburg plant where the wheel-gate leakage has been found to vary between 10 and 20 sec-ft. on a 4500-kw. unit under 210 ft. head. This is an automatic plant and the turbine inlet valve, which is of the plunger type, has been arranged to close automatically whenever the turbine gates are closed.

39 The turbine inlet valves used at the lower ends of the Davis Bridge penstocks are of the ordinary butterfly type, furnished by the manufacturer of the turbines. They are provided with bronze seat and snap rings to increase tightness and are controlled

by hydraulic cylinders operating on penstock pressure. They have motor-operated pilot valves so that they can, if necessary, be closed or opened from the switchboard. The cylinders controlling these valves are so piped that a valve can be operated from another penstock in case its own line is drained, which is a feature of considerable convenience for inspection, etc.

40 The valves at the top of the penstocks are shown in Fig. 13. They are of the same size and design as those at the intake, except that a single threaded stem is used instead of the stem and connecting rod adopted for the intake valves. The hoist, illustrated

FIG. 13 DAVIS BRIDGE PENSTOCK AND AIR-VENT VALVES

in Fig. 14, is mounted on trunnions which permit it to follow the angular movement of the stem as the valve operates, tightness at the point where the stem passes through the body being obtained by means of a ball-and-socket joint provided with a cup leather. The valves are designed to close under free spouting velocity due to the full head of 160 ft. and are remote-controlled so that they can be closed from the power house in case of accident, the time required to close being about $3\frac{1}{2}$ min. The remote-control feature was not adopted for opening as it was not considered safe or desirable to fill the penstocks from a distance, since this might be done too rapidly if the valve opening were not carefully watched. No bypasses are used with these valves. To insure maximum dependability direct-current hoist motors were adopted, so wired that if necessary they can be operated from the station batteries, which would probably be available if all other sources of power should fail. Ordinary operation is from the exciter system.

The hoists are of rugged design and are provided with roller thrust bearings instead of the ball bearings commonly used. This is standard New England Power practice for all heavy hoists,

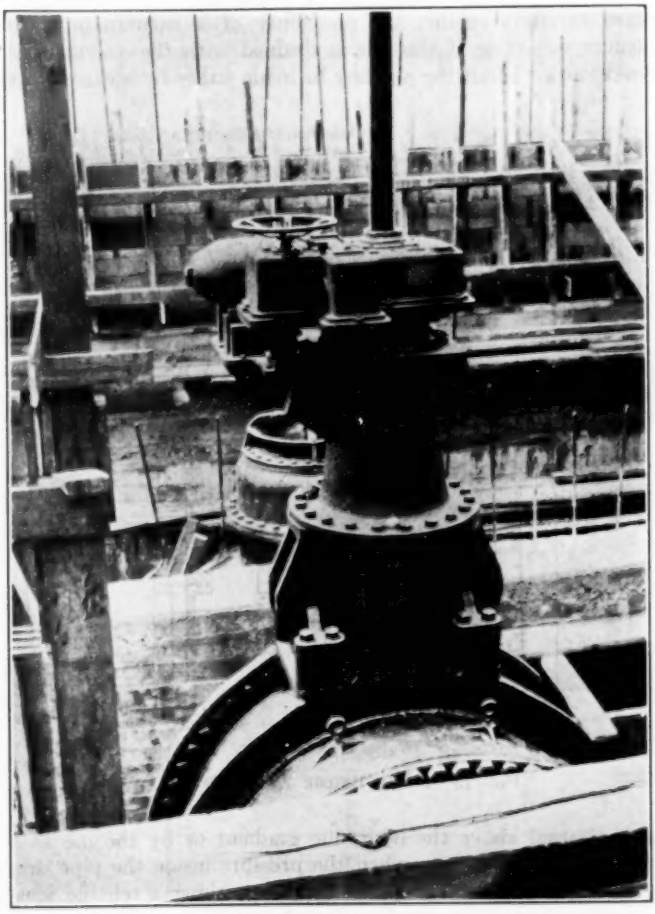


FIG. 14 HOISTING MECHANISM, DAVIS BRIDGE PENSTOCK VALVE

experience having shown that the roller type is much to be preferred for such service. Like the valves at the intake the upper penstock valves are provided with babbitt seats and are extremely tight. The leakage at 150 ft. head does not exceed 2 gal. per min. each.

Air Valves and Vents

41 One of the commonest penstock accidents is collapse due to internal pressures below atmospheric and whenever valves are installed at the upstream end of a pipeline it is necessary to guard carefully against the possibility of a vacuum or partial vacuum occurring if the line is drained with the valves closed. Provision for admitting air may be made either by means of open

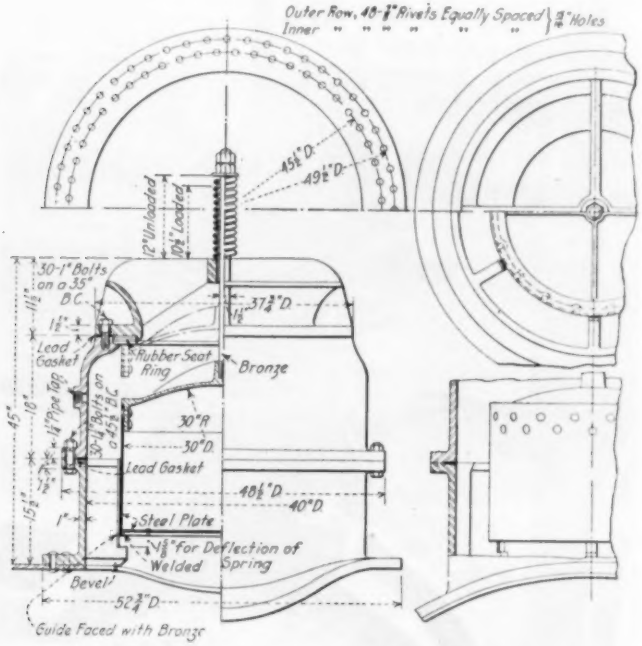


Fig. 15 Davis Bridge Air-Vent Valve

vents carried above the hydraulic gradient or by the use of air valves designed to open when the pressure inside the pipe drops below atmospheric. The problem of furnishing a reliable means of venting is far from simple and each installation must be studied in the light of local conditions. Ice is without doubt the greatest danger. Both vents and air valves must be carefully protected from cold and in most cases artificial heating is required to prevent freezing. Vents should be capable of admitting air at least fast enough to replace water drawn off through the turbine, and this without an undue drop in pressure to draw it in. In some cases, for instance with wood-stave pipes, a comparatively small vacuum

is sufficient to cause trouble. If a turbine gate is suddenly opened with the penstock full and the headgate closed it may be impossible for the water column in a high vent pipe to accelerate quickly

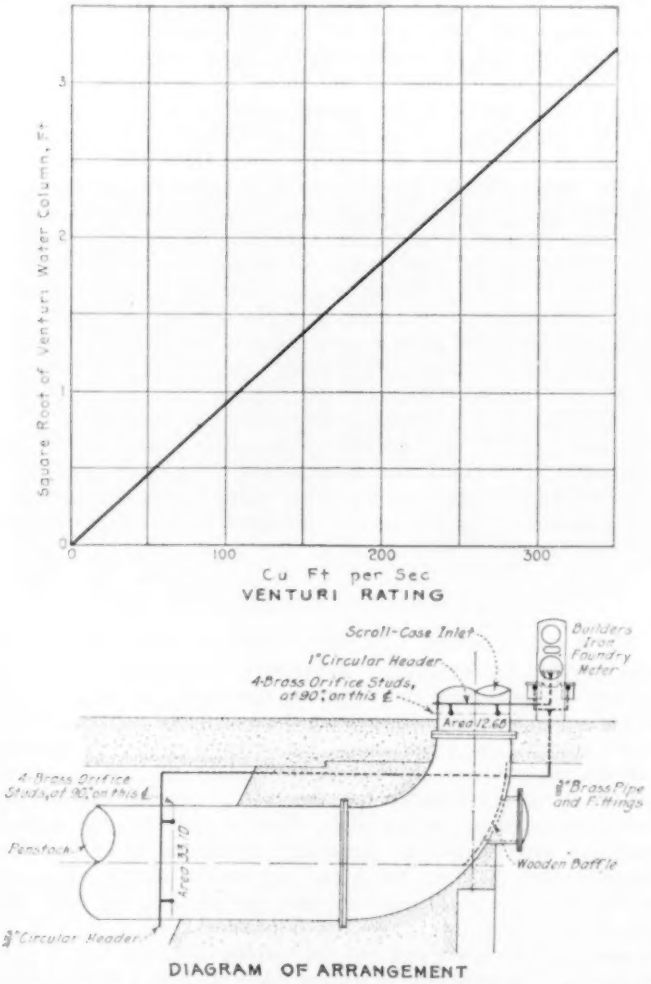


FIG. 16 DIAGRAM AND CALIBRATION OF PLANT NO. 5 VENTURI INSTALLATION

enough to prevent the formation of a serious vacuum, in which case the penstock would probably collapse before the vent emptied of water so that air could flow in. If a vent valve is used the possibility of sticking seats, etc., must be considered.

42 With one exception, vents where required on the New England Power System are of the open type. Below the Davis Bridge penstock valves, however, such vents would have had to be nearly 200 ft. high to avoid overflow under load changes, and while these could have been run up and supported on the surge tank, the question of accelerating the water column and of preventing freezing resulted in the choice of air valves. For this service valves having a 24-in.-diameter opening were developed and installed immediately below the penstock valves. They are shown in Fig. 15, and, as will be seen, are closed by a float which rises as the penstocks fill. The bottom of the float is perforated to prevent collapse under the high pressure and also to reduce shock on closure by permitting the water to rise in the float and compress the entrapped air. The float is of bronze and rides loosely in bronze-faced guides. When closed a flat machined seat on the float bears against a ring packing in the housing. This ring is of hard grade and is made of a composition of rubber and duck. The valve is tight and shows no particular tendency to stick, though it opens with considerable suddenness when the pressure in the penstock drops. The opening is cushioned by the spring above the yoke at the top. All three valves are housed in a single compartment provided with doors hinged to swing in either direction to admit or emit air when the valves operate. Freezing is easily prevented by the use of a few electric space heaters in the compartment.

Discharge-Measuring and Recording Devices

43 A paper dealing with the economical operation of hydro plants would be inexcusably incomplete if it failed to touch on devices for measuring and recording discharge. Nothing is more important to economy of operation than complete and trustworthy water records and a convenient and reliable method of checking turbine performance. In a high-head plant the use of venturi meters is usually possible but the cost is considerable and the head loss by no means negligible. Several years ago the author's company investigated the feasibility of calibrating an inherited pressure drop at Plant No. 5 and the results, shown graphically in Fig. 16, were so satisfactory that the practice has been continued in later plants. The No. 5 penstocks are 6 ft. 6 in. in diameter at the lower end and the turbine inlets are 4 ft. 0 in. in diameter. The reduction in section takes place in a 90-deg. bend. Venturi connections were made at the inlet and outlet ends of this bend, and by means of a mercury U-tube the pressure drop was determined at various discharges, the flow being measured by pitometer traverses. It was found that at full gate a pressure difference of slightly over 10 ft. of water was available and that this difference varied as a straight-line function of the square

of the velocity. In other words, the pressure followed the regular venturi law and the constant was practically identical with that for a standard venturi. No difficulty was encountered in obtaining a venturi indicator-register-recorder instrument calibrated to suit the constant determined and this was installed as shown in Fig. 17. This metering device was described in *Engineering News-Record*, March 2, 1922, and is thought to be the first installation of its kind.

44 The same expedient has been used at the Searsburg and Davis Bridge plants, the drop used in the first case being from

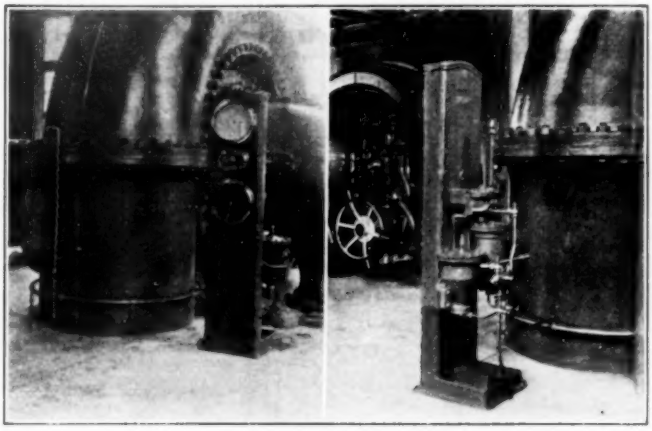
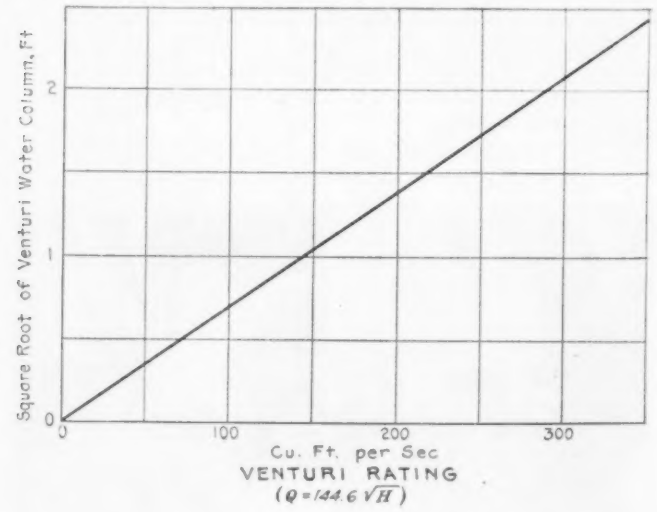


FIG. 17 PLANT NO. 5 VENTURI INSTRUMENT

the 6-ft. 6-in. inlet to the 4-ft. 8-in. outlet of the plunger valve, in the second from the 9-ft. 0-in. diameter penstock to the most constricted section of the 7-ft. 0-in. butterfly valve. The calibrations in both cases were made in connection with turbine tests by Prof. C. M. Allen, the water being measured by the salt velocity method. The results of these calibrations are shown in Figs. 18 and 19, which also show in outline the valves and connections.

45 At Searsburg the connections are such that when the plunger valve is closed the upstream connection in subjected to full pressure, while the downstream pressure drops to zero. This would cause the meter to register full discharge with the turbine shut down, but the difficulty is overcome by the use of electrically operated valves in the pipes leading to the instrument. These valves are tied in with the automatic control of the plant, and when the plunger valve starts to close they operate in such a way as to equalize the pressures on the two sides of the instrument and cause it to register zero. At Davis Bridge the connections are both on the upstream side of the butterfly when this is closed, and no complications are introduced.

46 In order to eliminate the danger of change in calibration due to the formation of tubercles, etc. near the venturi connections,



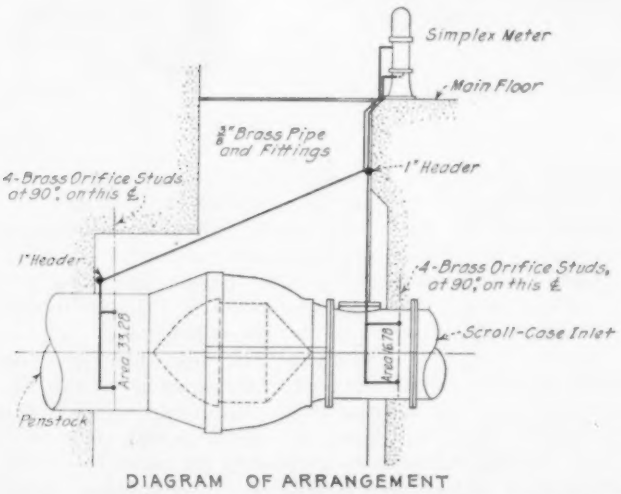


FIG. 18 DIAGRAM AND CALIBRATION OF SEARSBURG VENTURI INSTALLATION

these connections are made by the use of brass studs with faces about two inches in diameter set flush with the inside walls of the waterway. This reduces the danger of change of constant to the

chance that the pressure drop between the upstream and downstream venturi connections will change, but since by far the greater part of this drop is due to the increase in velocity and only a small fraction to friction dependent on the condition of the pipe surface, the probability of a serious variation of constant is remote. No

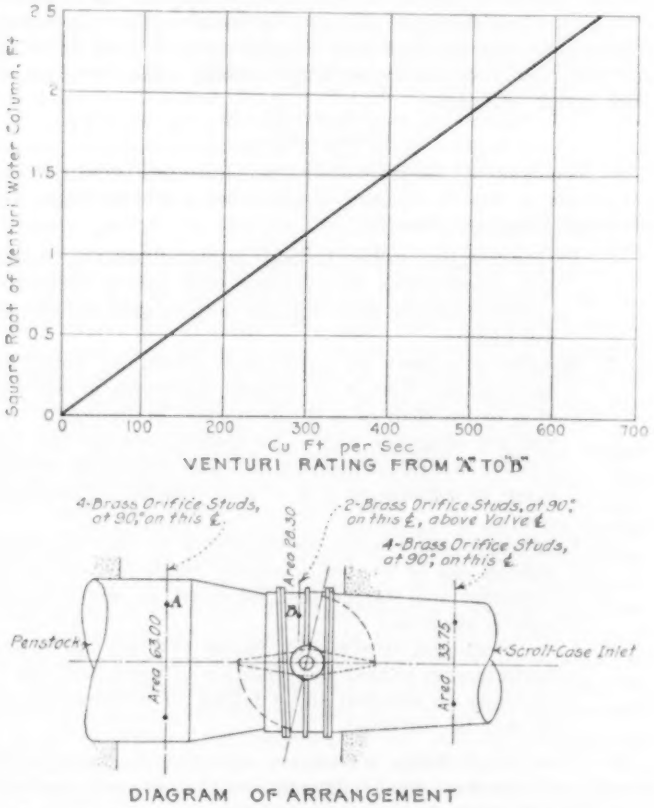


FIG. 19 DIAGRAM AND CALIBRATION OF DAVIS BRIDGE VENTURI INSTALLATION

evidence of a tendency to vary has been noted in the Plant No. 5 installation which has been in service since 1921.

47 The average low-head plant does not afford any similar pressure drop which can be used for permanent water measurement. Any head loss, however, such as that through trash racks, intake gates, etc., can be calibrated for temporary use, and while this calibration will generally not hold for long it is likely to be very useful for the duration of a test, since it affords a means of

taking a great number of observations in a limited time. Probably even more important is the possibility it offers of determining the characteristics of a turbine without any actual measurement of water whatever. On the assumption (apparently always correct) that the discharge is in proportion to the square root of the pressure drop, it is possible to determine the shape of the efficiency curve of a wheel and plot, say, gate or power against percentage of maximum efficiency, and have a perfectly good guide as to the best range for operation, even though nothing whatever is known about actual efficiencies.

Conclusion

48 In conclusion, the author wishes to state his belief that for best results as regards economy of operation in a hydro plant, the following things are essential:

1 Proper design to reduce leakage and head losses to reasonable values. This does not mean, of course, that more money should be spent for this purpose than the savings will justify

2 Suitable provisions for rapid and convenient inspection and repairs. This is even more important in the case of a hydro than of a steam station. A hydraulic turbine out of service with water spilling over the dam and a market for power means a total loss. If a steam turbine is out of commission it is at least possible to economize coal

3 Continuous and reliable water records so that operating policies can be fixed with a clear comprehension of the discharge and storage that has been, is, and probably will be, available

4 Convenient and trustworthy means of checking turbine performance, so that the condition of equipment can at all times be watched and a falling off in efficiency can at once be corrected.

49 Given these things, economical operation becomes merely a matter of exercising good judgment and the necessary vigilance to see that the desired policy is carried out.

50 The reliability of a hydro installation is largely determined before it ever goes into service. It demands suitable equipment, selected with a proper regard to the service required, the use of rugged designs but with careful attention to vital details, and the avoidance of complications. The author believes that strength and simplicity are the most important essentials of any kind of hydraulic equipment, and these qualities are the keynotes of American design. Most of our troubles come when these ideals are forgotten or disregarded.

DISCUSSION

H. L. Doolittle.[1] The use of valves at both the upper and lower ends of the penstock is certainly correct. There are few cases where a valve at one end only would be justified. In western practice, where the heads are very high and penstocks are sometimes over a mile long, double valves are imperative. It is not considered good practice to empty the penstock except for inspection or repairs. The filling of a long penstock requires considerable time in order to avoid the entrainment of air or severe surges. It is therefore essential to have a valve at the bottom of the line, that can be closed whenever the water wheel is out of service. This is particularly true with turbine installations, in order to avoid wear on the turbine gates.

Remote control for opening penstock valves should never be used. Our practice is to install remote control for *closing* the valves at the top and the bottom of the line, but they can be opened at the valve only. Cases have been known where loss of life has occurred due to the opening of the upper valve before the penstock was reported clear for operation. When a penstock is undergoing repairs, such accidents as this can be prevented by locking the valve and giving the key to the man in charge of the repairs. It is then certain that the valve cannot be opened by mistake.

It is gratifying to see the increasing interest in the use of permanent water-measuring devices for hydroelectric plants. For many years great amounts of money have been expended on indicating and recording instruments for steam plants, but similar equipment was not seriously considered for hydroelectric plants. The maintenance of the prime mover at its highest possible efficiency is just as important in a hydroelectric plant as in a steam plant, and one of the greatest aids to this result is some form of meter that will give a continuous record of the water supplied to the wheel. During the last three or four years the Southern California Edison Company has installed venturi meters on the units in all of its principal hydroelectric plants, and has adopted this as standard practice for all new installations.

In one stream-flow plant, where the water must be used without storage, the crews on different shifts compete with each other to see which one can turn out the maximum number of kilowatt-hours. The venturi meter is of great assistance in this case, of course, and its intelligent use has resulted in an increased output from this plant. In another high-head impulse-wheel plant a continuous check on the wheel efficiency enables the operators to determine when needles and nozzles have become worn so that they can be replaced before a serious loss of output occurs. It is an actual fact that needle and nozzle tips are now changed much more fre-

[1] Assistant Construction Engineer, Southern California Edison Company, Los Angeles, Cal. Mem. A.S.M.E.

quently than before the meters were installed, with a consequent saving of several hundred dollars monthly.

Some objection has been raised to venturi meters, due to the loss of head in the tube. In a very low-head plant this may be a serious factor and the author's suggestion to use a taper section of pipe or a reducing elbow may be a very ingenious way to overcome this trouble in such cases. The tube friction loss of 1½ to 2 ft. in a high-head plant is of no great importance. It is true that this friction head may represent a loss of power valued at five or six hundred dollars a year, but the benefits derived from the intelligent and continual use of the meter would result in savings that would offset this loss many times.

JOHN F. VAUGHAN.[1] In discussing the relation of design to operation, the author emphasizes the necessity for safety and brings out the desirability of eliminating as far as possible the element of human uncertainty — the danger of dependence on the acts of the operator. To make a device "foolproof," whether it be for the control of flood waters on a river or for governing the flow of traffic on a railroad, reads all right, but let us not fool ourselves into a state of overdependence on "automatic" operation. A flood gate that fails wide open; a signal system that in failing can indicate only danger — they fulfill their functions; but a method or device that relieves the operator from the necessity for eternal vigilance and healthy responsibility, and by failure to operate can produce a hazardous condition, by so doing may actually increase rather than eliminate the hazard.

What seems to be a tendency in some of the best of our recent hydraulic designs today to depend more on human action even in the control of floods and the protection of life and property, may be right or wrong. At least it gives food for thought. For illustration: Some years ago the writer was directed to equip certain sidings and secondary trackage in a terminal yard of a third-rail system with cut-out switches. He refused to do so on the ground that a third rail known to be always alive may be safer than one supposed to be normally dead. One day an old lady found her way into the yard and started across the whole group of tracks, stepping from service to third rail and from third to service rail. Her shoes were dry and the soles sewed, and she got across, but the order for the cut-out switches was cancelled. Even from a commercialistic point of view this problem may be of the utmost importance and should be given the most careful consideration. There are limits to dependence on "automatic" devices for the relief of human responsibility.

[1] Vaughan Engineers, Boston, Mass. Mem. A.S.M.E.

O. H. BUNDY.[1] The author has made a valuable contribution to the engineering design of the headworks of hydroelectric developments, especially in his statements of the requirements of flashboards, crest gates, headgates, and operating mechanism. His frank statement of difficulties encountered in operating sluice gates is particularly commendable, especially so as the proceedings of the engineering societies are filled principally with descriptions of successful accomplishments. In many cases engineers have to go through the same experiments that others have tried and found unsuccessful before they obtain the proper answer.

The design requirements of reliability, practicability, and minimum economic cost must always be kept before the designing engineer of hydroelectric developments. Due to the increased cost of coal, starting about 1916, many projected hydroelectric developments began to show a net saving in total costs over steam-produced energy. In the last three years there has been a marked advance in the overall efficiency and economy of producing electric energy by steam, so that the increased cost of fuel has been largely neutralized in some sections. If our national resources are to be economically conserved by the maximum development of available water power, it is necessary for the designer to use the utmost skill in the selection of type and detail design of the component parts of any development. Ingenuity and vision must be used to the utmost to produce the final desired answer: i.e., lowest economical cost per kw-hr. produced.

In Par. 15 the author refers to the installation of six 72-in. disk-arm valves for discharge sluices in the dam of the Sturgeon Pool plant of the United Hudson Electric Corporation. A statement of the operating experiences with these valves may be helpful.

Leakage. The six valves were the first installed using the disk-arm principle. They are equipped with bronze seats. Later designs use babbitt seats, which, to the writer, is a material improvement. Three tests have been run to determine leakage with results as follows:

Valve No.	Sept. 24, 1924	Oct. 30, 1924	April 13, 1925
1	0.277	0.0415	0.0106
2	0.292	0.076	0.0072
3	0.960	0.083	0.0142
4	0.228	0.0396	0.0011
5	0.635	0.0607	0.0100
6	0.503	0.0810	0.0100
Total sec-ft.....	2.895	0.3818	0.0531
Average sec-ft	0.4825	0.0636	0.0089
Gal. per min., total...	1300	172	24

The decrease in leakage is accounted for, we believe, by the gradual seating of the valve, which probably takes a little longer

[1] Engineer, United Hudson Electric Corporation, Poughkeepsie, N. Y.

than with babbitt seats. The valves have been operated from time to time and we have not accounted for the reduction in leakage by silt filling in against the upstream face.

Erosion. The original design of the discharge conduit, shown in Fig. 20, included a transition section from a circle with a diameter of 72 in. to a square section 6 ft. by 6 ft. After the dam was completed in November, 1923, the valves were operated practically continuously for a period of about two months to control the elevation of the reservoir, and at frequent intervals for the next

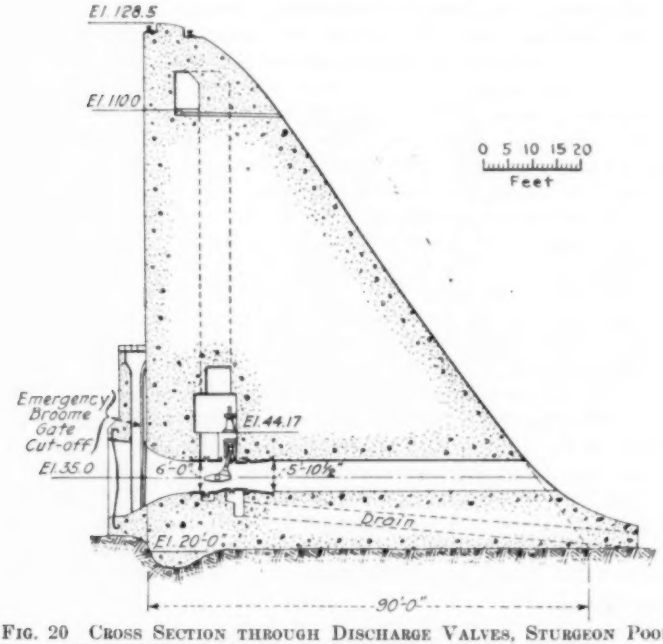


FIG. 20 CROSS SECTION THROUGH DISCHARGE VALVES, STURGEON POOL DEVELOPMENT, UNITED HUDSON ELECTRIC CORPORATION

three months to control the spillway discharge. At the end of this period an examination of the conduits disclosed severe erosion, especially in the transition section. Consideration was given to several methods of repair and it was decided to excavate sufficient concrete to obtain a good bond and replace with a circular section 72 in. in diameter. A dense concrete proportioned about 1: 1½: 3 was used with wood forms, covered with a steel lining. The result was a smooth, even surface, offering a minimum of rough surface where cavitation could start. The valves have been used only a few times since the repairs were made, but there is no indication of erosion. The aging of the concrete should increase the resistance,

and the writer believes the use of a bitumastic surface coating is desirable.

The discharge of water under high head with velocities of from 75 to 100 ft. per sec. requires a careful design of the valve, the

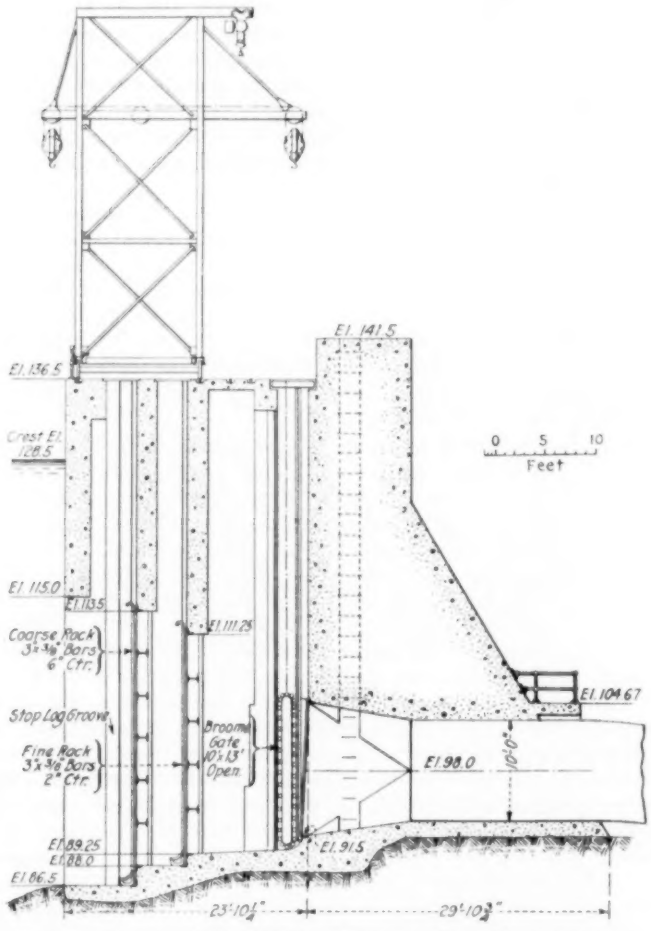


FIG. 21 INTAKE CROSS-SECTION, STURGEON POOL DEVELOPMENT, UNITED HUDSON ELECTRIC CORPORATION

mechanism, and conduits. The extensive experience of the U. S. Reclamation Service with high velocities has been gathered together in two volumes under the title of High Pressure Outlets, by J. M. Gaylord and J. L. Savage. The subject-matter is excellent and

well worth consideration by any one contemplating the design of similar facilities for handling high-velocity discharge.

Location. The author's statement in Par. 16 that for higher heads and free discharge to air the control valve should, if possible, be located at the downstream end of the water passage, should be underscored. The writer's experience indicates that this is of major importance in securing successful operation, as an opportunity is afforded to secure maximum venting of the valve to eliminate vibration and it exposes only a minimum of surface where erosion can take place. The use of renewable rings, possibly of flaring cast-iron-flange construction, is indicated. In many cases designing difficulty may be experienced due to the necessary location of the valve in the spillway section proper.

The writer notes with interest the remarks of the author on the design of racks and accessories, and takes this opportunity of briefly describing the intake of the Sturgeon Pool Development of the United Hudson Electric Corporation. During the preliminary investigation of this development it was found advisable to make a careful study of the intake construction due to the difficulties experienced at other plants. After reviewing the results obtained in a number of different installations of similar conditions the following conclusions were reached:

1 In practically all of the plants where the racks extended to or near the surface of the water, extra labor was required and troubles were experienced, sometimes shutting down the entire plant or one or more units during runs of leaves, brush, and frazil ice.

2 Practically all plants where the water is taken in from areas ten feet or more below the surface and with large open pondage above the intake experienced a minimum of trouble, the leaves and débris riding on or just below the surface, and the frazil ice disappearing in the pond.

The general design of the intake was developed as shown in Fig. 21. The features indicated as desirable were incorporated in the final design by the J. G. White Engineering Company. It consists essentially of three bays, one for each 10-ft. penstock. The water required is 600 sec-ft. for each unit at full load giving the following velocities:

	Ft. per sec.
Coarse rack section	1.5
Fine rack section	1.9
Broome gate section	4.6
Penstock	7.6

No measurements have as yet been made to determine the difference in head between the forebay and bell mouth.

The forebay cut-off wall is carried down 13.5 ft. below normal pond level. A stoplog groove is provided for access to the rack structure. The racks are removable with a tripping basket on the

bottom to catch trash falling off the rack as it is hoisted. Experience indicates that only one rack is required, probably with a 2½-in. spacing.

For each penstock there is a Broome gate. The gates, including an emergency gate for closing the up-stream entrance of the discharge valves, are handled by a motor-driven traveling hoist.

The intake has now been in operation for seventeen months and the racks have been cleaned only twice. The leaves and small brush in both cleanings filled only about three wheelbarrows, requiring a minimum of extra labor for operation. The surface water at the intake is free from eddies and is often partially covered with leaves and debris, and there is no indication that 1800 sec-ft. of water is being drawn into the intake below the surface.

HERMAN WILL.[1] This paper covers plants where a penstock is used. The Vernon plant, for instance, does not come in this class. In several hydroelectric plants not using penstocks, an instrument has been used which continuously records the percentage of gate opening of the turbine. This instrument is simple in construction, consisting of a circular or strip chart operated by clock movement, the marking arm being pivoted and connected to a lever arm which in turn is connected to the governor of the turbine by proper reducing mechanism. The motion of the governor is transmitted through the lever to the marking pen and the record is made on a chart graduated in percentage of gate opening. Where a constant head is used, the chart can be graduated in units of flow. A comparison of this record with that of the head water provides an interesting check on the efficient use of water. For instance, water going over a spillway or through Tainter gates, when the turbine gate opening is less than 100 per cent, indicates power available but not used.

F. R. BENSON.[2] In the past the hydraulic-power-plant designer has been prone to give the intake-rack arrangement less attention than its importance warrants. Thousands of dollars have been spent in attempting to raise the peak of hydraulic-turbine efficiency a fraction of one per cent, yet a partially clogged rack which may cut down the plant output several per cent is passed up as a necessary evil. Even today we are often called on for rack-raking equipment for a new plant only to find that the space allowed for the trash platform is not only insufficient for a mechanical rake, but is inadequate even for successful hand raking. This paper has been presented at a most opportune time and will help

[1] Sales and Service Engineer, The Bristol Company, Waterbury, Conn.

[2] Assistant Hydraulic Engineer, Newport News Shipbuilding and Drydock Company, Newport News, Va.

to fill an important gap in hydraulic practice on which there is little information available.

The value of the mechanical rake can no longer be questioned. In many plants today, one man with a mechanical rake is easily handling most severe runs of trash which formerly would have shut down the plant despite the efforts of thirty or forty men with hand rakes to clear the racks. The Vernon installation, to which the author refers, is the thirty-fifth mechanical rake which we have now in successful operation.

ROBERT W. ANGUS.[1] The discharge-measuring and -recording devices described by the author in Pars. 43 to 47 are a further illustration of the many uses to which the principle used in the venturi meter may be put, in a simple way. The author, however, shows straight-line relations in all cases between cubic feet per second and the square root of the drop in head. This differs from the general experience in such devices, for it means a constant coefficient for the meter. Experiments on venturi meters made by the writer, and those recorded by various experimenters, show that the coefficient increases with the throat velocity, at least until the latter reaches about 15 ft. per sec.

The writer computed the coefficients corresponding to the curves in Fig. 16, 18 and 19, basing the results on scaled values, in all but Fig. 18. Using the ordinary venturi-meter formula, the velocity at the throat in each of the three cases is

$$v = 8.69c\sqrt{H} \text{ in the elbow, Fig. 16,}$$
$$v = 9.30c\sqrt{H} \text{ in the Johnson valve, Fig. 18,}$$
$$v = 8.98c\sqrt{H} \text{ in the throttle valve, Fig. 19.}$$

These values correspond to coefficients c for the three cases of 0.990, 0.927 and 1.05 respectively, the last of which is clearly too high, and the first would appear to be high also.

Using these coefficients the loss of head h between the two gage connections may be computed from the formula $h = (1-c^2)H$, and is found to be

$$h = 0.02\ H = 0.0027v^2 \text{ for the elbow}$$
$$h = 0.141H = 0.0019v^2 \text{ for the Johnson valve.}$$

The formula for the elbows shows a loss much less than that indicated by experiments, and would confirm the conclusion that the coefficient in this case is too high. While the writer has no information about the loss in the Johnson valve, the loss in it figured from the author's tests appears to be smaller than would be expected. This method of checking the results, taken along with Professor Allen's measurements, would enable a result to be obtained that is more accurate than that used by the author.

[1] Professor of Mechanical Engineering, University of Toronto, Toronto, Ontario, Canada. Mem. A.S.M.E.

That the venturi-meter principle may be used with a fair degree of accuracy in very abnormal cases was demonstrated by the writer some little time ago in connection with a 6-in. by 3-in. venturi meter in which a block of wood had lodged. This block was of such shape that it jammed well up in the throat of the meter, and naturally very seriously interfered with the flow. The meter was submitted to the writer for rating with the obstruction in place, and it was found that even under this condition, the coefficients had a variation of only about 10 per cent between the full capacity and about one-fifth of the full capacity, so that it served as a fairly reliable meter even under such a very bad condition. In this particular case the coefficient decreased as the velocity increased.

THE AUTHOR. The discussion which this paper has brought forth is gratifying as an evidence that in spite of its rather general and discursive character, it may contain material of some interest.

Mr. Doolittle's comments bring out in a number of respects the similarity of eastern and western practice. In the two New England power plants most nearly resembling those of which Mr. Doolittle speaks, considerable study was given to the penstock-valve problem before valves at both ends of the lines were adopted. It is an interesting confirmation of the soundness of the decision arrived at to know that in the West double valves are considered imperative. A similar correspondence is to be noted in the practice of using remote control for *closing* but not for *opening* penstock valves, as well as in the sentiment in favor of some means of continuously metering water. It cannot be too strongly emphasized that water-measuring equipment if properly used will pay for itself many times over in increased output. In fact, it is indispensable to intelligent operation. The same can be said of turbine testing in general. It is simply folly to make the tremendous investments involved in hydroelectric developments and then guess at the best method of operation. The few hundreds or thousands of dollars necessary to completely determine the operating characteristics of the turbines will inevitably bring a greater return than any other equal expenditure. An increase of one or two per cent in the average annual output of a plant, which can very reasonably be expected from more intelligent operation, may look trifling, but it means a corresponding increase in the gross revenue from the plant and a much greater percentage increase in the net income. It may make all the difference between a common stock which pays a generous dividend and one which pays nothing at all.

Mr. Vaughan apparently gathered from the paper that the author favored a generous use of automatic devices to eliminate the element of human uncertainty. This is far from the impression that it was intended to convey. Throughout, emphasis was laid upon the desirability of the maximum simplicity as conducive to

the maximum reliability. Automatic equipment in general involves increased complication, and to that extent is likely to reduce reliability. The author has consistently opposed the use of alarm devices etc. which tend to relieve the operator from the necessity for the " eternal vigilance and healthy responsibility " which Mr. Vaughan rightly prizes. In advocating foolproof designs the author has merely had in mind equipment and structures which could not be wrecked or cause serious damage in case an operator made a mistake or failed to perform his duties. Opposition to automatic devices should not be carried to an extreme, however, as there are many cases in which they are helpful and desirable. The numerous automatic plants in satisfactory service have demonstrated that automatic equipment, if properly designed and maintained, is entirely reliable and practicable.

From a theoretical standpoint, Professor Angus' criticism of the straight-line calibrations shown for the various venturi installations is no doubt justified. As a matter of fact, however, the tests indicated that the error involved in using an unvarying discharge coefficient throughout the operating ranges would be inappreciable. It certainly does not exceed a very small fraction of one per cent. In Par. 47, however, is a statement to the effect that the discharge is always proportional to the square root of the pressure drop, which the author must modify. Since the paper was written, two instances have arisen where this condition did not hold. Both of these involved pressures taken at various points in turbine cases, the connections being so located that impact and eddy currents might affect the differentials. In both of these instances the test results plotted up as straight lines on logarithmic paper, but the exponent of the discharge was found to be other than 2. However, under the comparatively simple hydraulic conditions prevailing in installations such as those described in the paper, a very slight error is to be expected from the assumption that the discharge is proportional to the square root of the pressure drop.

Professor Angus criticizes the calibrations because of the discharge coefficients of 0.990, 0.927, and 1.05 which he calculates from the data presented. If such coefficients were obtained in the process of calibrating a water passage approximating the standard venturi meter, the water measurements would certainly be open to question. In the instances cited, however, the author does not believe that these coefficients afford a proper basis for criticism. In the first place, these calculated values are based on the areas given for the various waterway cross-sections, which are probably not dependable within one or two per cent. A slight error in the assumed cross-section at the throat will make a large difference in the calculated coefficient of a venturi. The coefficient of 1.05 obtained in the calibration of the butterfly valve is based on a theoretical throat area taken from the manufacturer's blueprint. The unmachined disk casting might very easily depart enough

from the theoretical thickness to account for the impossible coefficient. In all cases under discussion, moreover, it is quite probable that impact and similar effects at the orifices tend to cause deviations from the venturi deflections which might be expected. In the case of the butterfly valve, the throat pressures are taken in the upper half of the valve only and it is easily conceivable that some influence, such as a vertical bend in the penstock upstream from the valve, might tend to cause a higher velocity above the wicket than below or vice versa, thus affecting the coefficient in a manner which could not be predicted. Lest this explanation confuse the issue, however, it should be clearly stated that the accuracy of the calibration is in no way dependent on the cross-sectional areas or the coefficients. No use whatever is made of these quantities, and they are of interest merely in calculating the losses through the devices calibrated. In practice, the venturi deflection, which can be determined with much greater precision than the areas of the waterways, is noted under various discharges as determined by any suitable method of measuring flow, and an instrument is installed which will indicate the correct flow for the pressure differences obtaining. The author therefore takes exception to Professor Angus' statement that checking by the calculated coefficients affords a means of obtaining results more accurate than that used.

Mr. Bundy's discussion forms a very valuable addition to the paper because of the detailed information he gives regarding the experiences of the United Hudson Electric Corporation. In connection with the tabulation presented showing the decrease in leakage through the Sturgeon Pool sluice valves, it is of interest to note that the valves when first put into service were practically tight. The leakage developed after a period of months, and by all indications was due to a gradual distortion of the valve housings resulting from shrinkage or compression of the concrete in which the inlet pipes are embedded. The condition is similar to that recorded by Gaylord and Savage in their book on High Pressure Reservoir Outlets in connection with the valves in the Elephant Butte Dam, and suggests that special provision should be made to avoid the possibility of distortion due to gradual compression of the surrounding concrete over a fairly long period of time after pouring, especially when valves are located in the lower parts of a high mass concrete structure. The description of the repairs to the discharge conduits of the sluice valves is of great interest and the results of further experience with these water passages will be extremely instructive. The author is very glad to have Mr. Bundy's enthusiastic approval in advocating the location, whenever possible, of free-discharge valves at the outlet end of the water passages.

The author has never felt that a gate-opening recorder such as described by Mr. Will was sufficiently good for the purpose of

water measurement to justify a relaxation of effort to find something better. In most high-head plants some venturi effect can be calibrated, so that the gate-opening recorder would be attractive in any case only for the low-head installations. These are precisely the installations in which the greatest percentage of head fluctuation is found, and, as Mr. Will intimates, the gate opening must be used in conjunction with the head in order to arrive at flow. In other words, the instrument could rarely be depended upon to record flow directly and a lengthy and cumbersome process would be necessary to work up actual discharge from the records obtained. Turbine-gate mechanisms are subject to considerable wear, so that in the course of a year the gate openings would probably change considerably without a corresponding change showing up in the records, and every time the turbines were overhauled the calibration of the instruments would have to be revised. In all modern plants complete records of kilowatt-hour output are kept, and the author believes that the flow obtained by working back from the output records by means of calibration curves would, in general, be more reliable and more easily calculated than those obtained by the use of the gate-opening recorded. The device would, however, be of some value as a check on operation, to show whether or not the turbines were working at the gate of maximum efficiency as consistently as possible.

Mr. Benson refers to the mechanical rack rake installed at the Vernon station. The author is glad to be able to report that this rake has amply justified itself and is a most valuable adjunct to the plant. Having had experience with it through one season's high water, the Vernon operating forces are enthusiastic in its praise.

The author wishes to thank the engineers who have entered into the discussion of this paper, both for the time and effort expended and for the valuable material presented.

No. 1967

MECHANICAL PROBLEMS OF HYDRAULIC-TURBINE DESIGN

By William Monroe White,[1] Milwaukee, Wis.
Member of the Society

Having determined the general characteristics of a hydraulic turbine to suit a given set of hydraulic conditions, the various mechanical problems arising must then be solved. These relate to the casing, which may be either of concrete, steel plate, or a casting; the guide vanes, with reference to their shape, support, clearance, and material of construction; the runner, its shaft and bearings; the governing mechanism; etc. The solution of these problems is indicated by the methods used in the design of the 70,000-hp. unit recently put in service at the plant of the Niagara Falls Power Company, Niagara Falls, N. Y.

THE art of hydraulic-turbine design in America received its greatest impetus upon the putting into operation of the first large turbines at Niagara Falls. Previous to that time such men as Francis had laid an excellent foundation for an American type of development as contrasted to what had been done previously in Europe.

2 The part taken by American manufacturers in the further development and perfection of hydraulic-turbine apparatus can probably be best evidenced by the statement that during the last twenty-four years there has been expended under the author's direction alone possibly a half-million dollars solely for development, invention, experimentation, and test of apparatus forming a part of what today is termed "the hydraulic turbine and its equipment." Notwithstanding this, there is yet much to be done, although we have rather definitely outlined the constants fixing the water passages to, through, and from the turbine, and have fairly definitely fixed the types of auxiliary apparatus to provide for the convenient, economical, and safe operation of the turbine.

[1] Manager and Chief Engineer, Hydraulic Department, Allis-Chalmers Mfg. Co.

Contributed by the Power Division and presented at the Spring Meeting, Milwaukee, Wis., May 18 to 21, 1925, of The American Society of Mechanical Engineers.

3 Assuming, then, the general characteristics of a given turbine to have been determined from well-developed formulas, it is proposed here to deal with the mechanical problems of hydraulic-turbine design, taking up in order problems connected with the flow of the water through the casing, the guide vanes, the runner,

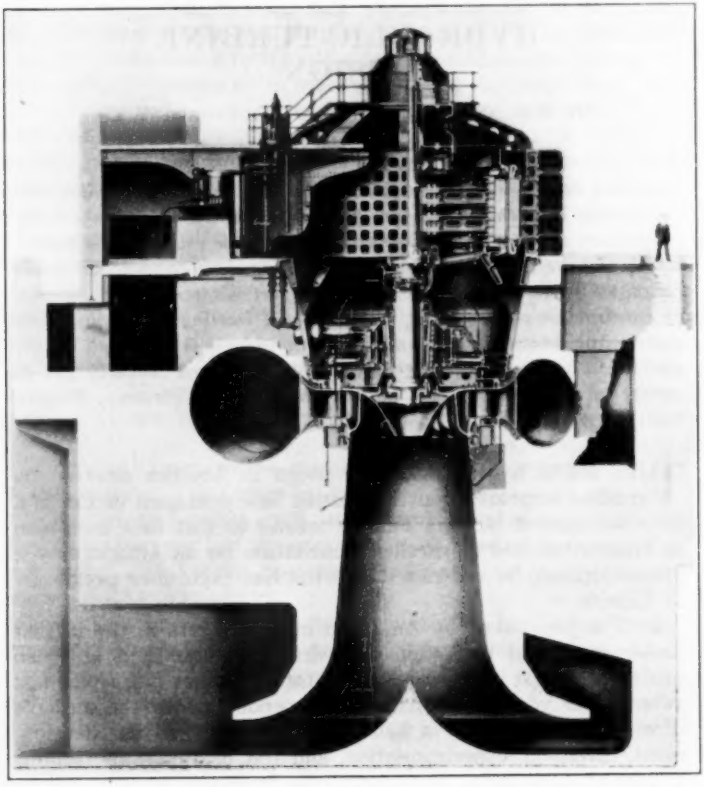


FIG. 1 COMBINED HYDROELECTRIC UNIT RATED AT 70,000 HP., 65,000 KVA., NIAGARA FALLS POWER CO.

(213 ft. head, 107 r.p.m. Plate-steel spiral casing, concrete hydraucone, direct-connected flyballs, adjustable lignum-vitae bearing.)

the draft tube, and then the mechanical transmission of power and its governing.

4 We can probably best discuss the matter by taking a concrete example such as the 70,000-hp. turbine recently put into operation at the plant of the Niagara Falls Power Company, Niagara Falls, New York. A cross-section through the turbine and generator of this unit, which operates under 213 ft. head at 107

r.p.m., is shown in Fig. 1. The first mechanical problem is to provide for the delivery of the water from the supply pipe to the runner. For this purpose a plate-steel circular-section spiral casing, Fig. 2, was used. It is shown in more detail with its speed ring in

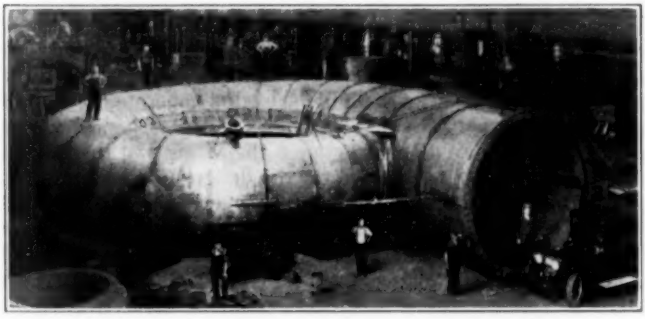


FIG. 2 PLATE-STEEL SPIRAL CASING FOR 70,000-HP. TURBINE

Fig. 3. In figuring the stresses for the casing one must consider both the normal operating head and the maximum head due to increase of pressure in the casing caused by the inertia of the water when it is stopped by the governor's rapidly closing the gates of

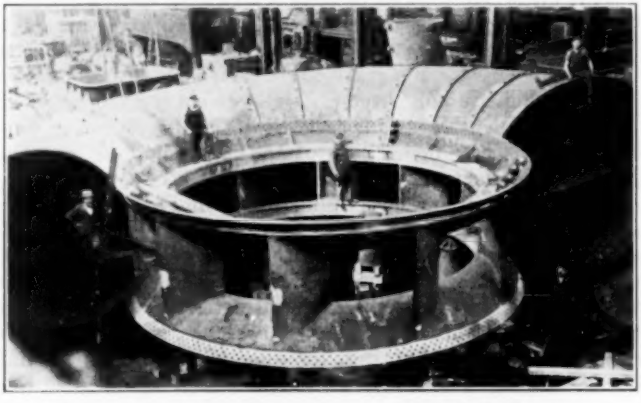


FIG. 3 CAST-STEEL SPEED RING FOR 70,000-HP. TURBINE
(Made in five sections and held together with bolts and shrink links.)

the turbine for speed regulation. Provision must be made not for the static pressure alone, but also for the shock resulting from this surge, which in some cases approaches water hammer. Thicker metal must also be used to provide for possible deterioration

caused by rust and erosion on the interior and on the outside of the casing. For this purpose $\frac{1}{8}$ in. has usually been added to the computed proper thickness of the plates.

The Casing

5 Examination made of penstocks and plate work in service for twenty years and over shows that there is a surprisingly small amount of deterioration due to rust and erosion. Plate-steel spiral casings, as in Fig. 2, are usually imbedded in concrete. It has been found advisable not to paint the outside of the casing, as a firmer bond will thereby be secured between the casing and the concrete.

6 The greatest problem met in the design of the spiral casing for the 70,000-hp. unit at Niagara Falls was the driving of the $1\frac{3}{8}$-in. rivets, $6\frac{3}{4}$ in. long, through the flange of the speed ring and through the double thickness of plate, and also the driving of the $1\frac{3}{8}$-in. rivets at the lap joints of the larger section of the casing where the plates are $1\frac{1}{4}$ in. thick. To drive these rivets a bull riveter was developed, and was constructed by the Hanna Engineering Works. This machine is capable of exerting 150 tons with 100 lb. air pressure and has a reach of 108 in. It is supported on a single cable and operates in any direction in order to accommodate itself to the various angles of the rivets about the speed ring and the lap joints of the casing. The riveter is adjusted to any required angle by means of two air motors, one of which rotates the jaws of the riveter and the other varies the angularity of the center line of the reach of the riveter. Fig. 4 shows this riveter in action. On actual test it drove sixty $1\frac{1}{4}$-in. rivets per hour, which is undoubtedly at a greater rate than could be accomplished on the usual fixed bull riveter. These rivets were of such size that it was thought inexpedient to attempt to drive them by hand. This riveter is now doing duty at the plate shops of the Allis-Chalmers Manufacturing Company's plant.

7 The plates of the steel spiral casing in Fig. 2 are made of varying thickness from the large diameter around to the smaller section of the spiral, providing for a uniform stress of nearly 7000 lb. per sq. in., taking into consideration the efficiency of the joints. They are $1\frac{1}{4}$ in. thick at the large diameter, decreasing to $\frac{5}{8}$ in. at the small section. Flange steel is used as material for most of the plates, and firebox steel for some of those requiring more difficult flanging. The interior of the plate-steel casing is not quite as smooth as that of the cast-steel, but its diameter is made larger, providing for sufficiently lower velocities to more than outweigh this difference. This can readily be done because of the lighter weight and lower cost of the plate-steel casing as compared with cast-steel casing.

8 A cast-steel speed ring illustrated in Fig. 3 serves to support the inner ends of the plates of the casing. This speed ring is made in five sections, all of cast steel, each section weighing 28,000 lb.

9 It is no great accomplishment to compute the stresses in the steel casting, given the sizes and loads; the real mechanical problem lies in so shaping the design that it will have the least initial strains when cast. It will be noted in Fig. 3 that each of the five sections has two ribs or vanes. The upper and lower flanges are extremely heavy, but having only two vanes tying the two flanges together, any difference in the contraction of one vane on cooling will not produce stresses in the other vane; but should there be three vanes in each of the sections, the unequal cooling of any one of the three vanes would necessarily bring stress upon the others. In order to be assured of homogeneous metal at the

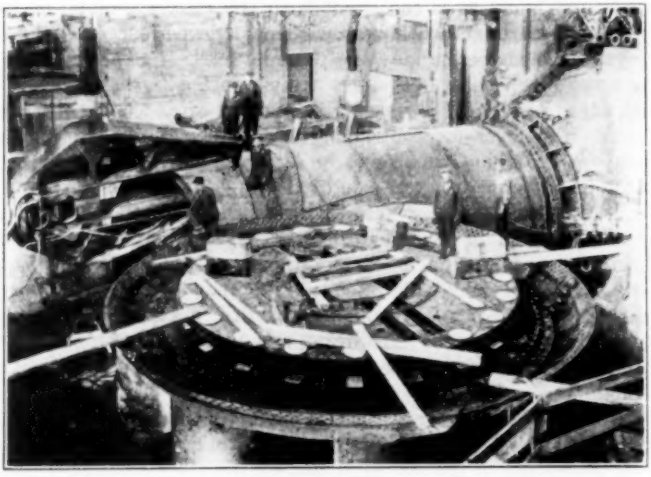


FIG. 4 150-TON, 108-IN. REACH, ADJUSTABLE BULL RIVETER
(Capable of driving sixty 1⅛-in. rivets per hour on spiral casing for 70,000-hp. turbine.)

junction of the vanes and the flanges or crowns of the speed ring, it is advisable to drill a pilot hole to the center of the mass at the junction of the vane and crown of the speed ring for examination. It will be noted in Fig. 3 that the sections of the speed ring are bolted together by heavy bolts; in addition a shrink link has also been found necessary to hold together the inner portion of each joint. A speed ring is a critical member in the design of any hydraulic turbine because the full flow of the water must pass between the ribs, which must be set at such an angle as to accommodate the flow with least disturbance and friction and yet be of sufficient thickness and shape to take the greatest strain imposed upon any part of the turbine. A wrong angle of the speed-ring ribs, or a wrong contour, may produce a disturbance

in the flow which will extend through the guide vanes, through the runner, and into the draft tube. Any great disturbance in the flow at the speed-ring rib may tend to cause serious pitting of the runner.

Guide Vanes

10 The control of the water flow to the turbine introduces probably the greatest mechanical problem in its design. The almost universal method now in use is to place from 16 to 24 wicket vanes, or guide vanes, on a circumference just inside of the speed ring, and to use an outside type of gate gear where the closed spiral is used. The diameter of the guide-vane bolt circle fixes the size and, consequently, the cost of the turbine, so that this dimension is made as small as is consistent with the proper guiding of the water to the runner with best efficiency. Experience has shown that a wedge-shaped guide vane provides for best efficiency. With the outside-type gear the vanes are supported by lugs or pivots cast at each side of the vane, with one lug extended for the connection of a lever for controlling the position of the vane. The stems on the guide vane must be large enough to resist the full water pressure against one side of the vane under full maximum-head conditions with a bearing pressure allowable for proper lubrication, and of a diameter such that the torsional stress is within safe limits under all operating conditions. The bearings supporting the guide-vane stems are bronze-bushed, each lower bearing being lubricated through a hole drilled axially through the stem. Individual grease lubrication, now preferably of the Alemite type, has been found to be most suitable and convenient for the attendants.

11 The clearance between the edges of the guide vanes and edges of the adjoining distributor plates is fixed by the considerations of leakage on the one hand, and on the other by the necessity for sufficient clearance to minimize friction for governor operation. Too close clearances should not be attempted because the distributor plates at the ends of the vanes do rust as well as the ends of the guide vanes, decreasing the provided clearance.

12 Fig. 5 shows a partial section through a modern high-head Francis turbine for 350 ft. head. It will be noted that the stems of the guide vanes are extended and an outboard bearing provided near the end of the stem to take the forces produced by the actions of the operating lever, or guide-vane lever, secured to the end of the vane. A stuffing box should be provided at the exit of the stem from the cover to prevent leakage. This prevents the washing of the grease from the bearing as well as dirt from getting in, and although many attempts have been made to avoid this type of construction, it has been proved that it is not a useless luxury but is worth its additional cost. The guide vanes are all moved by the motion of one shifting ring, through suitable links, levers, and pins. The guide vanes are machined where each vane touches

the next, and when new the leakage at the points of contact between the vanes is practically nothing, because the clearance in a modern turbine does not exceed a few thousandths of an inch. The length of the lever arms of the guide-vane stems and their relation to adjacent parts should be such that when the link is broken the outer end of the guide vane cannot swing into contact with the outer edge of the runner vanes. In the Allis-Chalmers Manufacturing Company's design the relation of the guide-vane lever with

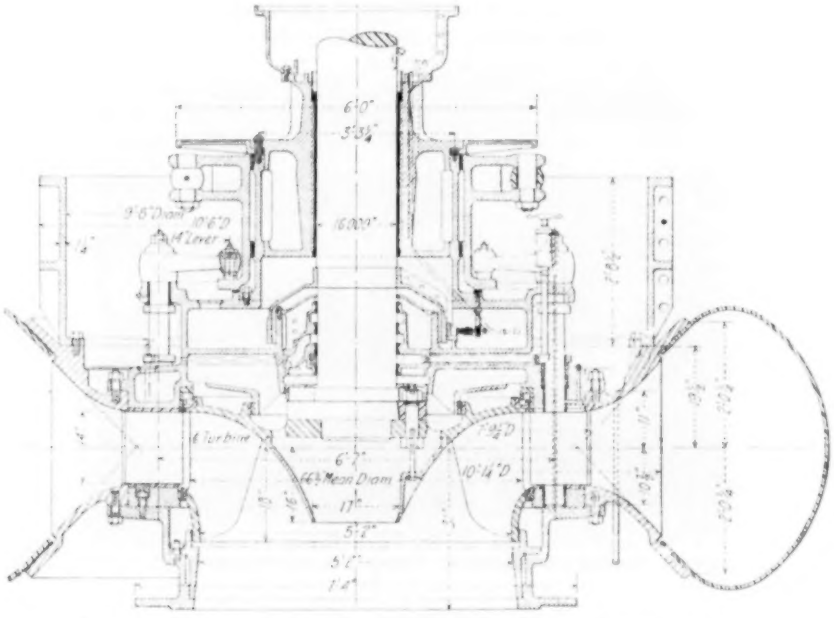


FIG. 5 SECTION OF 20,000-HP., 350-FT. HEAD, TURBINE, SHOWING OIL-LUBRICATED BEARING, VISCOSITY PUMP, GUIDE-VANE THRUST BEARINGS, AND RENEWABLE LINER BELOW RUNNER. SPEED, 360 R.P.M.

reference to the adjacent parts is such that the large end of the guide vane will not swing and touch the runner vanes.

13 For accuracy of adjustment of a vane with reference to an adjacent one, an eccentric pin is used either on the shifting ring or at the end of the guide-vane lever, whereby slight adjustments can be made. This adjustment is also useful in case the stem of a guide vane becomes sprung due to some foreign substance getting between any two guide vanes. A breaking point is now provided between the shifting ring and each guide vane whereby a part will be ruptured in case a foreign object is caught between two vanes. The shape of the guide vanes, the force of the governor, and the

yield point of the breaking connection all have to be carefully and properly related to secure on the one hand instant relief under undue stress, and on the other hand freedom from too frequent and unnecessary interruptions.

14 Thrust bearings are provided on guide-vane bearings for thrust or weight in either direction axially of the vane, and adjustments are convenient for keeping the guide vane in position for equal clearances between the end of the vane and its adjacent distributor plate. In the case of vertical turbines for moderate and low heads, where the weight of the vane is greater than the upward thrust of the water pressure on the area of the guide-vane stem, the upward-thrust bearing is omitted.

15 Cast-steel guide vanes are universally used for moderate- and high-head plants having outside gate mechanism, but for low-head plants a plate-steel guide vane has been developed which is

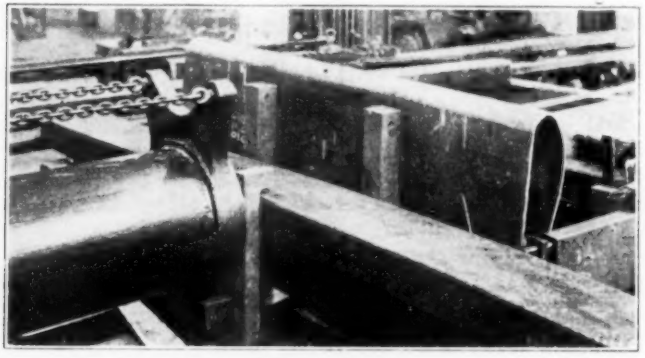


FIG. 6 84-IN. PLATE-STEEL GUIDE VANE BEING TESTED TO DESTRUCTION TO DETERMINE STRENGTH. USED IN MANY PLANTS UP TO 70 FT. HEAD

not only economical but even more suitable than the cast-steel because of the flexibility and malleability of the plate which permit it to be brought back to proper shape when distorted. Fig. 6 shows an 84-in. steel-plate guide vane under cross-bending test in a hydraulic press to determine ultimate strength and point of failure.

16 The velocity of the water as it discharges through the guide vanes is about as high as the velocity of flow through any part of the turbine. The cutting action of sand or other foreign substance in the water greatly increases with increased velocity, so that the parts adjacent to the ends of the guide vanes are usually faced with easily renewable plate-steel surfaces to provide for taking care of this wear without great expense. The clearance between the edges of the upper and lower crowns of the runner and their adjacent matching surfaces affects materially the efficiency of high-head units, and these clearances must be maintained at a

minimum for maximum efficiency. For this reason it is advisable to have wearing rings around the periphery of the upper and lower crowns of the runner, and also renewable wearing rings in the adjoining places in the upper and lower crowns of the distributor. It is difficult to hold the wearing rings on the circumferences of the upper and lower crowns of the runner. The most satisfactory method is not only to shrink the rings on to the runner, but also to fasten them with screws as well. Frequently the rings are heated to such a temperature that they may be slipped over a shoulder and shrunk into a recess turned on the runner, which makes an excellent job but one that must be very carefully done. For

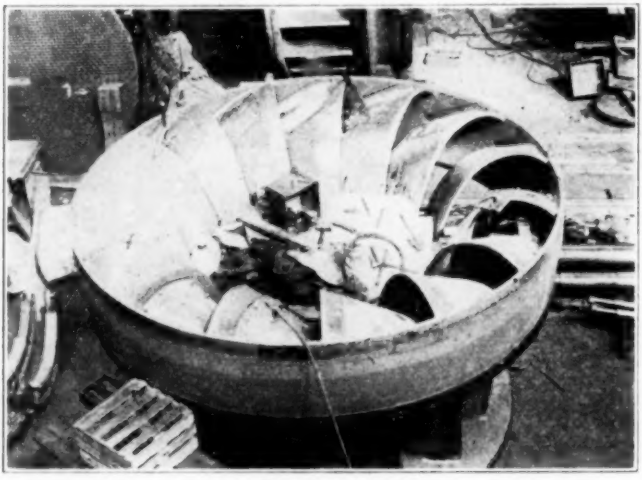


FIG. 7 RENEWABLE PITTING PLATES AT DISCHARGE EDGE OF RUNNER
(One of eight 45,000-hp. runners for Duke Price Power Co.)

medium and low heads where the water conditions are permissible these wearing rings are omitted.

THE RUNNER AND ITS SHAFT AND BEARINGS

17 The turbine illustrated in Fig. 1 operates under a head of 213 ft. or a pressure of about 100 lb. per sq. in. This pressure on the large surfaces of such a turbine produces forces that are of astounding proportions. It is indeed a mechanical problem to cause proper flow through the runner and to provide runner vanes which will withstand the tremendous forces resulting under operating conditions. It is a simple matter to compute the forces acting on a vane of the runner due to the difference of pressure from one side of the vane to the other which would be caused by the velocity of the water under smooth streamline-flow conditions, but

the water does not flow under such conditions, and the factor by which to multiply the computed stresses for arriving at the probable stresses is one which is obtained only as a result of long experience. That region below the discharge edge of the runner vanes under abnormal conditions of head, and sometimes under normal conditions, may be subject to tremendous forces caused by eddies and vortices, and as these eddies and whirls break up and the vortices collapse, internal water-hammer conditions are caused which may rupture any cones or supports or mechanical plates placed in that region. The rumbling and low vibrations of turbines such as sometimes occur in normal practice arise from this action of the water. In some plants this may not be noticeable, but in

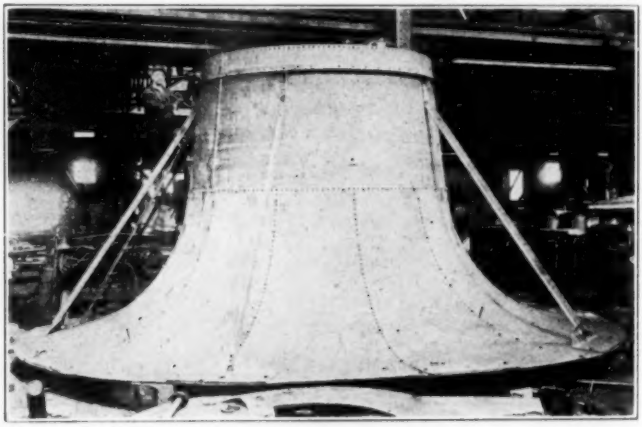


FIG. 8 PLATE-STEEL HYDRAUCONE ELIMINATES COMPLICATED FORMS AND SPEEDS UP CONSTRUCTION

others it is accentuated because the natural period of some structure may correspond closely to that of the disturbance.

18 In order to provide for repairs to pitted runners a method has been worked out as illustrated in Fig. 7. This improvement consists in casting depressions in the vanes on the discharge side near the small opening to the next vane and setting therein steel plates held to the body of the vane by rivets, thus providing means for easy removal. Due to the almost mechanical action of the vortices at the discharge of the runner, it has been found advisable in many cases to install wearing plates on the draft tube near the discharge from the runner as illustrated in Fig. 5.

19 When a large hydraucone such as that shown in Fig. 1 is constructed of concrete, the form work is expensive and costly. A plate-steel hydraucone as illustrated in Fig. 8 has been used quite extensively and has been found economical and a great time saver in the construction of the substructure of the power house.

20 The enormous forces brought on the runner by reason of the collapsing of the vortices above mentioned produce strains in all directions. The runner, however, must be held rigidly in a central position because of the close clearance around its periphery. Consequently not only a strong but a stiff shaft is required. A shaft of nominal diameter, that is, one computed to take care of the torsional stresses only, could be made stiff by employing nickel steel, chrome-vanadium steel, or manganese steel, but stiffness can best be obtained by increase of diameter. Hence it is mechanically better to make the shaft of large diameter, using ordinary tough carbon steel instead of using a shaft of smaller diameter of the extra-quality material, either shaft costing about the same.

21 One of the important mechanical problems is the design of a proper guide bearing for the main shaft in the vicinity of the runner. Lignum-vitae has been used for many years with considerable success. The tendency at the present time, however, is to the use of oil-lubricated, babbitted guide bearings. With this latter type of bearing satisfactory lubrication can only be accomplished, first, by preventing the oil from being drawn out of the reservoir by suction of the air along the shaft to the runner, and second, by preventing the water which leaks out of the same stuffing box, under conditions of high tailwater and high load, from getting into the oil reservoir supplying this bearing. The oil-lubricated, babbitted guide bearing is particularly advisable where the generator is so closely coupled to the water wheel that only one common guide bearing is used for the support of both the water-wheel runner and the water-wheel end of the shaft of the rotor of the generator. On 60-cycle generators the clearance between the poles and the yoke is rather small, and it is necessary to keep the clearance between the shaft and the bearing within close working limits.

22 The tendency of recent designs is to make each vertical guide bearing and its oiling system self-contained. Gear pumps are used in some cases to circulate the oil within the individual bearing. Another efficient device is the viscosity pump illustrated in Fig. 5, which provides for this lubrication. Still another arrangement is that shown in Fig. 9, where a flexible member such as a chain or spring passes around the shaft above the bearing and is led over pulleys at a right angle and down outside of the bearing into the oil reservoir below the bearing. Such a device was thoroughly tried out at the Green Island Plant of the Ford Motor Company.

23 The 70,000-hp. unit at Niagara Falls, although of the two-bearing type, has lignum-vitae bearings, but the water in the Niagara River is so clear that serious wear does not occur and small clearances can be maintained. However, those bearings were made adjustable to provide for the maintaining of close clearances after the lignum-vitae had worn away, and in addition a large settling basin was used at Niagara Falls which provided unusually clear water.

24 Practically all of the lignum-vitae bearings built during recent years have been of the adjustable type. A very satisfactory bearing of this type consists of an outer housing and four or more cast bearing blocks designed to fit into the housing, the inner surface of the blocks being lined with strips of lignum-vitae bored out to form the bearing surface. Bronze strips are provided where

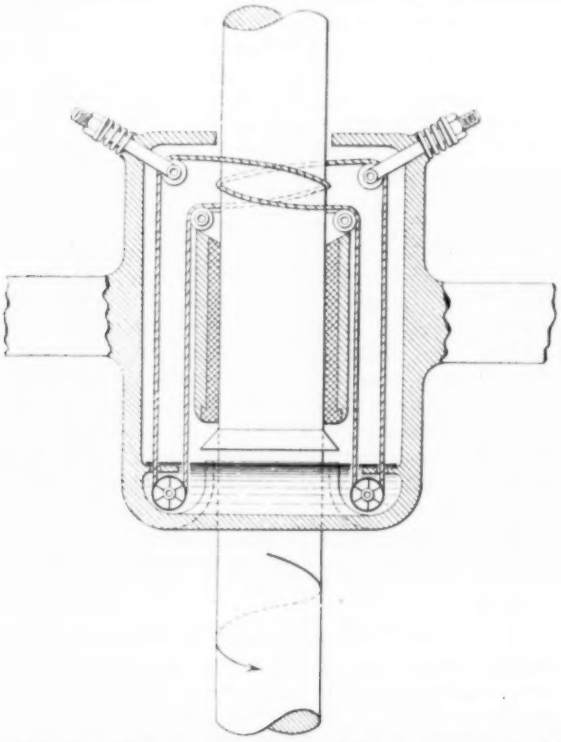


FIG. 9. FLEXIBLE SPRING DRIVEN BY SHAFT BRINGS CONTINUOUS SUPPLY OF FRESH OIL TO TOP OF BEARING. USED ON GREEN ISLAND TURBINES

the blocks rest in the housing, and in case of wear shims may be inserted under these strips so as to bring the bearing surface out to its original position. By using shims of different thicknesses the blocks may be adjusted to any position to compensate for unequal wear.

25 Another type of adjustable bearing quite similar in construction may be adjusted by sliding the bearing blocks down farther into the housing guides, these being tapered. Still another type has the bearing blocks supported by eccentric or radial bolts, which provide for adjustment.

The Governing Mechanism

26 The tremendous power of the modern hydroelectric units necessitates a governing mechanism of great delicacy, because the slightest change of the gate mechanism means considerable change in the power output from the unit. The mechanical problem involved in securing this result is the providing of a positive and smooth drive for the flyball mechanism, since the perfection of governing is primarily dependent upon the fixed, continuous, and definite ratio of the speed of the flyballs with respect to the shaft and the speed of the unit. It has long seemed to the author that belts and gears have no place for the driving of the flyballs of large hydroelectric units, and to obviate the necessity of these a directly connected flyball has been developed which is mounted directly upon the main shaft. Such a device is illustrated in Fig. 1. This device has proved so delicate and so positive in its action that it has been incorporated in practically all of the hydroelectric units manufactured by the Allis-Chalmers Manufacturing Company during the past six years. Delicacy of motion is secured by providing ball bearings at all moving points of the flyball. Positive motion of the valve is secured by using large flyball weights. This provides for such exact and positive gate motion that the gates are not constantly moved by a jiggling valve, but are only positioned when a change of load demands a new position of the gates. By using this direct-connected flyball device wear and tear on the gate mechanism is reduced to a minimum. Oil has been found to be decidedly the best means for furnishing the fluid pressure for governor operation.

27 Recently a number of plants have been placed in operation where no governors are used, the gates of the unit being locked and only shifted when the load despatcher notifies the operator that the output of his station is to be changed. These units feed current into large systems where some of the units in other power houses are equipped with governors. Before the days of the large power systems, governors were a necessity, because the load fluctuations frequently were a large percentage of the connected load. However, the percentage load fluctuations have not kept pace with the growth of transmission and power systems. A fluctuation of 1000 kw. on a 10,000-kw. unit was not considered extraordinary on an isolated system, but what 300,000-kw. system of today, except in case of abnormal trouble, receives an instantaneous load variation of 30,000 kw.? Probably an instantaneous change of 4000 or 5000 kw. represents the average fluctuation of the 300,000-kw. system of today, or a fluctuation of approximately one-sixth of that of the isolated system. The author feels that governing equipment will be simplified more and more, and that as a result many forced shutdowns now caused by failure of accessories and governing equipment will be eliminated, with an appreciable decrease in the

cost of operation and an increase in the total output per kilowatt of installed capacity.

28 At the upper end of the penstocks feeding the three 70,000-hp. units at Niagara Falls are located three 23-ft. 6-in.-diameter butterfly valves, one of which is shown in Fig. 10. These valves take the place of the customary type of sliding headgates, and in addition are designed to open against full pressure or close with full velocity with a pressure of 50 ft. head on the center of the valve. Neither of these extreme operating conditions can be met by the ordinary type of headgate used in the majority of present-day plants.

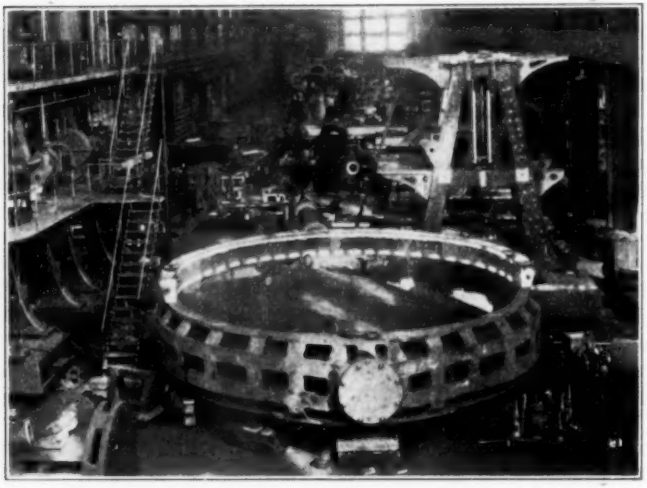


FIG. 10 LARGEST BUTTERFLY VALVE EVER BUILT BEING ASSEMBLED IN SHOP. DIAMETER 23 FT., 6 IN.; HEAD, 50 FT.

29 The pivot on which the 23-ft. 6-in. wicket is carried is vertical, and when in closed position is designed to withstand a total force of 1,250,000 lb., this being divided on two 19-in. trunnions. It being impossible to cast and ship this valve in one piece, the housing was made in six sections and the wicket in three sections, the wicket being 36 in. thick at the center section. The valve is operated by a hydraulic cylinder located 50 ft. above the valve and capable of moving the valve with 150 lb. oil pressure. This pressure is transmitted to the wicket through a 24-in.-diameter hollow shaft.

30 Butterfly valves may be made practically tight by inserting piston rings in the edge of the wicket. Such an arrangement is illustrated in Fig. 11, which shows a 96-in. valve designed to operate under 212 ft. head. Butterfly valves usually seat at an angle of

from 8 to 12 deg. from normal for closed position. One half of the wicket is therefore wedged tighter by the water pressure but the other half is deflected, and unless some form of sealing device is used, some leakage will occur with large valves under high heads. By cutting a groove around the periphery of the disk, a bronze piston ring may be inserted which is pressed out by water pressure and springs so that the leakage will be insignificant; on a 108-in.-diameter valve under 421 ft. head it was less than one-fifteenth of a cubic foot per second.

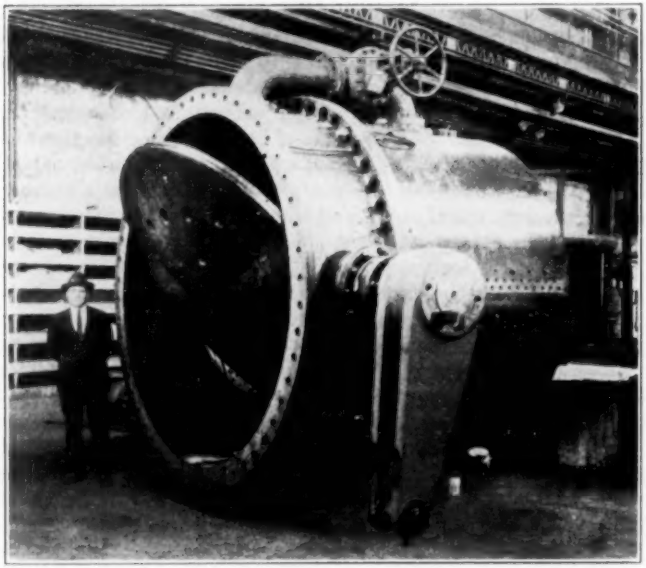


FIG. 11 96-IN. BUTTERFLY VALVE FOR 212 FT. HEAD. CAST-STEEL WICKET EQUIPPED WITH BRONZE PISTON RING DECREASES LEAKAGE

31 On large vertical hydroelectric units the problem of obtaining a proper fit between the flanges of the turbine and generator shafts is difficult. Take a 25-ft. shaft 24 in. in diameter with 42-in. flanges. Suppose there is a discrepancy of 0.003 in. in one flange. This may cause the shaft to throw as much as 0.060 in., which is entirely too much. Yet it is no simple matter to machine such a flange within 0.003 in., and rather than risk a long delay when coupling up in the field, the Allis-Chalmers Manufacturing Company make it regular practice, whenever possible, to bolt the two shafts together in the shop and turn them in a lathe to test for trueness before shipping. Any unevennesses which could be quickly remedied in the shop would require days of fitting in the field, with still a doubtful job.

32 It has not been the author's intention to try to cover all the mechanical problems to be met in the design of a hydraulic turbine, as that is not possible. Only by continual experimenting, studying, and improving can we make hydraulic turbines of 100,000 hp. capacity operate uninterruptedly in order to serve continuously the many industries and homes that consume the output of these units and rely upon them for their living.

DISCUSSION

H. Birchard Taylor.[1] The author mentions the use of a plate-steel casing in the 70,000-hp. unit which was built by his company for the Niagara Falls Power Company. The writer believes that in the field of moderate-head installations there are conditions which call for the use of plate-steel casings, and considers that the method of construction developed by the author has furnished an excellent solution of the casing problem in this field; namely, in that region where the forces are too great for the economical use of a concrete spiral casing and where they are not great enough to warrant the expense of a cast-steel or iron casing. On the other hand, for heads as high as that employed at Niagara Falls, namely, 214 ft., even when the casing is as large as those of the 70,000-hp. units, the writer believes that there is not sufficient difference in cost between cast steel and plate steel to warrant the use of the plate casing. In the two 70,000-hp. units built by the Cramp Company for the same plant, the writer proposed and adopted the use of cast-steel casings.

Some of the advantages of the cast over the plate casing are the speed, cheapness, and convenience of erection, the fact that it can be made absolutely tight against leakage, its great rigidity against deformation, and the assurance of long life. The difficulty of effective calking and the impossibility of locating leaks where the casing is imbedded to the concrete involve the risk of subjecting the surrounding concrete to high pressures when the installation is one under a high head. The life of plate casings is not known, nor is it known whether or not such an allowance as the $\frac{1}{8}$ in. additional thickness provided by the author for corrosion is sufficient. It is true that there are many examples of long life of plate work both in turbine casings and in plate-steel penstocks, but on the other hand there are examples where the life of plate work has not been so long. Since modern practice involves the burying of the turbine casing in the concrete substructure of the power house it would seem that in units of large size and of great importance, where the head is high, cast casings are to be preferred, and that their advantages will fully compensate for a considerable difference in first cost.

[1] Vice-President, William Cramp and Sons Ship and Engine Building Company, Philadelphia, Pa. President, Pelton Water Wheel Company. Mem. A.S.M.E.

Referring to the author's mention of the " necessity for sufficient clearance " between the guide vanes and the distributor plates to avoid friction and to provide allowance for rusting, it seems worth while to emphasize the importance of keeping this clearance a minimum, in order to avoid any considerable leakage flow during shutdown conditions, which is undesirable both on account of the loss of water and the resulting wear on the runner and on the guide vanes and the distributor plates themselves. The writer believes that close clearances should be used at this point with rugged construction of the surrounding parts to avoid binding due to distortion of the structure. Much of this binding in the operating mechanism is due to distortion of the turbine itself when under operating conditions. With rigid construction there is no doubt but that the clearances between the guide vanes and the surrounding parts should be made very fine in order to keep the leakage down.

An excellent means of reducing friction and leakage between guide vanes and distributor plates, and also for providing a strong connection between the guide-vane stems and the vanes, without the necessity of using unduly thick vanes, is the use of circular disks cast integrally with the vanes at top and bottom. A small circular portion of the distributor plate thus becomes a part of the vane and rotates with it. These disks thus block off a considerable part of the leakage space and provide an excellent construction, particularly for high-head, low-specific-speed turbines.

The author brings out an important point when he mentions that the operating gear should be so designed that in case of breakage of a link the tip of the free vane cannot come in contact with the runner. This is particularly important in the case of large units where racks have not been provided or where they must be lifted on account of ice conditions. Until recently this consideration was not generally appreciated, and while in some turbines of low and moderate specific speed the operating mechanism may have been so arranged that the breakage of a single link would result in the fouling of the vane lever with the remaining levers, preventing the forcing of the vane tip into contact with the runner, this condition was achieved by accident rather than by intention.

Moreover, this protection is not sufficient. If a single vane is freed by the breakage of a link, it tends to close to a part-gate position, and would move into the runner only when forced by an obstruction. If an obstruction enters the turbine it is not likely to limit its damage to a single guide vane or link but to be carried around with the runner and to break a number of consecutive links, or all of them. There are few low-specific-speed installations where the guide vanes would not then foul the runner vanes; and in turbines where cast-iron guide-vane levers are used, even the breakage of a single link might readily result in the fracture of the vane lever with resulting contact between guide vane and runner. One serious accident resulted from the use of too wide

a rack space, permitting the passage of logs of 6 in. diameter into the machine. Such wide rack spacing should not be used in any turbine, even if of very large size. There has been a feeling that as the bucket passages were large it was perfectly feasible to have a wide rack spacing. This is a fallacy. No unit will stand up against railroad ties or logs if they are permitted to come into the machine. The rack spacing should be kept conservative, the main reason being that the resisting power of an obstruction increases rapidly with an increase in size of obstruction and may assume values out of all proportion to the strength of the turbine.

In recent designs for low and moderate specific speeds, definite provision has been made by means of a torsional stop, limiting the angular motion of each guide vane to prevent contact between guide vanes and runner. This of course does not absolutely insure the runner or guide vanes against breakage if a large object should lodge between them, but it minimizes the danger.

In connection with the mention of the use of wearing plates on the runner at the seals between the revolving and stationary parts, the writer would call attention to the value of provisions for reducing the leakage in the first place, and thus minimize the wear caused by it. The use of either labyrinth seals or the recently developed rubber seal rings is of value in reducing this leakage and increasing the life of the sealing rings.

The author brings out an important point in calling attention to the severe eddies and vortices existing below the runner of the turbine. The writer agrees with his view that these vortices may cause tremendous forces with resulting severe water-hammer conditions in the top of the draft tube. There is no question but that an analysis of the flow at this point in the turbine will show the great value of filling up the central void below the runner by solid material, thus producing an annular draft-tube passage and eliminating the turbulence which originates at the axis.

It is not necessary to avoid any attempts to overcome the turbulence for fear that the structure may be too weak to serve the purpose, but rather it is necessary to provide a proper structure and thus to eliminate the cause of the disturbance and vibration. It is a simple matter to design central cones in the draft tube amply strong to carry any forces which may come upon them, as has been thoroughly demonstrated in many plants which have been in operation for long periods. There have been instances of draft-tube cones being cracked or distorted by the forces at this point, but in each instance it was found that reinforcement within the draft-tube cone was either entirely lacking or grossly inadequate, due to failure to provide proper overlap of the reinforcement bars at a construction joint.

A comparatively small amount of reinforcement steel properly placed will provide a wide margin for strength at this point. The writer believes that the provision of the central core below the

runner must be carried still further. The reason for the existence of shocks and turbulence is that we are attempting to discharge an annular stream of water from the runner which is unable to fill the central cavity on account of the fact that as the whirling water approaches the axis its velocity increases in inverse proportion to the radius and the distance to which it can approach the axis is thus limited. We therefore have discontinuity of flow, and the effect of a sudden enlargement of the passage, when the stream passes from the runner into an ordinary cylindrical draft tube. This point of transition from the runner to the draft tube is the only remaining point in the entire turbine waterway which requires correction to secure complete continuity of flow.

If continuity of passage area is provided so that the passage can run completely full at all points, vortices and turbulence can be avoided and absolute quietness of the draft tube may be secured. It is interesting to compare the earlier units at the Queenston Station and at the Niagara Falls Power Company's Station No. 3 with the latest units installed, namely, the 58,000-hp. units Nos. 6, 7, and 8 at Queenston and the 70,000-hp. I. P. Morris units at the Niagara Falls Power Company. These later units are all provided with complete continuous central cones in the draft tube and show a very marked gain in quietness of operation. These later units are remarkable for their quiet operation and freedom from any vibration whatever at any gate opening.

Supplementing the author's mention of the large butterfly valves installed as headgates for the 70,000-hp. turbines at Niagara, mention might also be made of the Johnson penstock valves with which all three units are equipped. These Johnson valves are believed to be a notable feature of the installation. They greatly contribute to the safety of the installation by providing an absolutely dependable means of completely closing the waterway. It will be recalled that at the time of the accident to the Ontario Power Station at Niagara the ability to close the Johnson valves under full spouting velocity undoubtedly saved a large part of the station from destruction; and there was recently occasion to close the valve on one of the 70,000-hp. units with the turbine gates open, which was accomplished quickly and without shock or other difficulty.

The author expresses the view that oil is the best fluid to use in the turbine governor system. The writer believes, however, that in the case of large installations there has been no finer governor system developed than that at the Cedars plant of the Montreal Light, Heat and Power Company, where water treated with potassium bichromate is used as the governor fluid and where one or two centrifugal pumps can supply the fluid for eighteen units. This system works without noise or any other difficulties, and has been in operation since 1913 with practically no wear of the governor, valves, or other parts. The central pumping system

arrangement permits the use of a small number of large pumping units instead of a separate pump for each turbine, and by thus increasing the size of these auxiliaries, reducing their number, and permitting their being located together, their convenience of operation and dependability are increased.

The author advocates machining the generator and turbine shafts as one shaft in order to insure proper alignment. The writer might state that in the building of several million horsepower of hydraulic turbines where the generators have been furnished by other manufacturers than the water-wheel builder, we have yet to hear of the first difficulty arising in the field due to improper connection between the generator and water-wheel shaft, or other difficulty due to their being separately machined. Moreover, in many hundreds of merchant and naval vessels furnished by the Cramp Company, the sections of propeller shafts have invariably been separately machined without the slightest difficulty ever having arisen from misalignment. The only thing required is high-grade mechanical work. The awkward and entirely unnecessary procedure of machining the shaft sections after assembling them appears both to the writer and to experts in shop practice whom he has consulted regarding this point to be entirely useless and unnecessary.

L. F. Harza.[1] Much emphasis has been laid by some manufacturers on the possibility of wide trash-rack spacing for high-speed units, on account of the large open spaces in the runner. As regulating gates must vary from zero to full opening during operation, and as the maximum opening of the regulating gates is far less than that between the blades of a high-speed runner, the writer would like to raise the question as to whether or not the regulating gates should not control the trash-rack spacing rather than the type of runner. A timber wedged in the regulating gates can cause shutdown of the unit, and it would seem that they should be protected by proper trash-rack spacing.

Forrest Nagler.[2] In the modern high-speed turbines, because of the larger angle at which water is delivered to the runner, the opening of the guide vanes is much greater than in the older style of wheel. Thus in an old-style 72-in. wheel, the opening would be about $4\frac{1}{2}$ in., while in the modern wheel it will be 8 or 9 in. An 8-in. obstruction of any length should not be passed through the wheel, for the reason that a piece of that size has sufficient mechanical strength to cause breakage of the gates. We would not recommend a rack opening of more than 6 in. on 72-in. wheels.

[1] Consulting Hydroelectric and Hydraulic Engineer, Chicago, Ill. Mem. A.S.M.E.

[2] Hydraulic Engineer, Allis-Chalmers Mfg. Co., Milwaukee, Wis. Mem. A.S.M.E.

No. 1968

TEST OF A UNIFLOW ENGINE

By Geo. H. Barrus,[1] Boston, Mass.

Member of the Society

Records of tests of uniflow engines are rare. The paper describes tests of a four-cylinder, vertical engine of 400-*kw. capacity at a speed of* 300 *r.p.m. Four series of tests were run at* 100, 200, 300, *and* 400 *kw., under the following conditions: saturated steam at* 125 *lb. pressure, atmospheric exhaust; saturated steam at* 125 *lb. pressure,* 5 *lb. back pressure; saturated steam at* 150 *lb. pressure, atmospheric exhaust; superheated steam at* 150 *lb. pressure, atmospheric exhaust.*

The steam consumption ranged from 20.7 *lb. per i.hp-hr. at* 125 *lb. pressure with atmospheric exhaust to* 15.3 *lb. per i.hp-hr. with superheated steam.*

THIS paper consists of a brief account of a series of tests which the author has recently conducted on a four-cylinder vertical uniflow engine, non-condensing, driving a 400-kw. a.c. Burke generator, direct connected, as shown in Fig. 1, with loads ranging from one-quarter to full load, with two different steam pressures, two different back pressures, two different percentages of clearance, and with saturated and superheated steam.

2 The tests were conducted for the Great Western Laundry Company, of Chicago, making use of the testing plant of the Ames Iron Works, Oswego, N. Y., in which the engine was designed and built.

3 The cylinders and heads are steam-jacketed. The cylinder jackets extend around a belt at both top and bottom of the cylinders with openings leading to the jacket spaces in each head. The steam on its way to the steam chest passes through the jackets, and the condensate, if any, is free to drop into the bottom heads, whence it is discharged through a mechanical trap. Except at very small loads the sweep of the steam keeps the condensate in motion, and the trap rarely discharges either water or steam.

4 The admission valves are of the double-beat poppet type, lifted vertically by a horizontal camshaft. This shaft is moved

[1] Consulting Steam Engineer.

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laterally by a horizontal centrifugal spring-resisted governor, which varies the cut-off according to the lateral position of the cams.

5 The cylinders are arranged in two pairs, with cranks 90 deg. apart. The two corresponding cranks of either pair are 180 deg. apart. Two cylinderfuls of steam are thus withdrawn from the source of supply at each quarter of a revolution. The total volume of these piston displacements is 0.71 cu. ft., that of the entire jacket spaces and steam chests is 8.2 cu. ft., and that of the throttle valve and piping from the cylinder flange to the stop valve on the

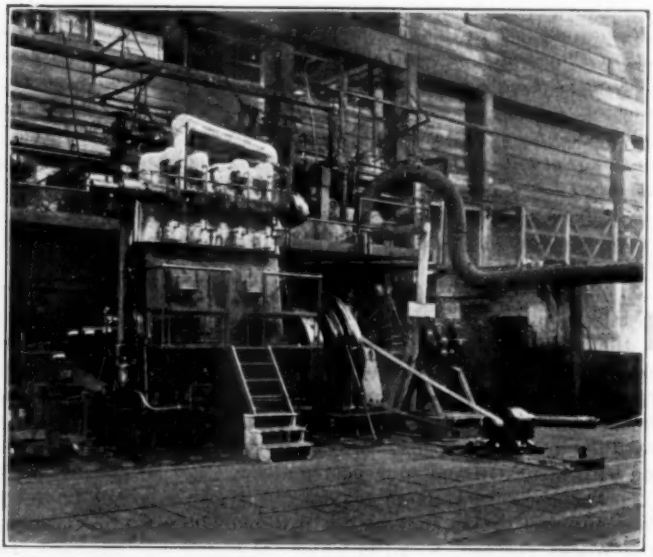


FIG. 1 UNIFLOW ENGINE FOR 400-KW. A.C. GENERATOR EQUIPPED FOR TESTING PURPOSES

steam drum, 2.0 cu. ft. These volumes are purposely mentioned to permit the comparison of the amount of steam withdrawn from the storage spaces at each double steam admission with the total space. The amount withdrawn each time is thus only about one-twelfth of the jacket volume. In this connection it was noticeable that the fluctuations of steam pressure at the throttle valve were extremely small; indeed, the cock on the steam gage at the throttle was kept wide open, and the gage pointer vibrated not more than its own width. The steam-pipe diagrams taken at the throttle valve revealed the same substantial absence of fluctuations, being practically straight lines. These results are obviously brought about by the influence of the multiple cylinders and the storage

spaces of the jackets. The practical absence of pressure variation in the steam main contributes not only to steam economy but to structural advantages as well, for it makes the pipe itself secure with a minimum amount of anchorage; further, it enables the size of the pipe to be reduced to the smallest permissible diameter — in this case 5-in. extra heavy. The velocity of the steam in the pipe at the greatest load (15,214 lb. per hr., average) is estimated at 108 ft. per sec. (6480 ft. per min.).

6 The moving parts of the engine are shut in by a casing, and no adjustment from the outside can be made, while in operation, except to vary the speed by tightening or loosening the governor spring. This construction admits of the efficient use of a pressure oiling system for the lubrication of the entire engine, no oil cups or grease cups being required.

7 The engine as equipped for testing purposes is shown in Fig. 1. Its dimensions are given in Table 1. The apparatus for measuring the steam consumption consists of a surface condenser of ample capacity, placed below the level of the engine and discharging by gravity into a pair of steel-plate weighing tanks, which are alternately filled and emptied.

8 In conducting the tests the methods recommended by the A.S.M.E. were substantially followed. The tests were carried out in a continuous series for each set of conditions, starting with the light load, then changing to the next load, and so on, or vice versa. Each test was of sufficient duration to collect a half-dozen or more tankfuls at a substantially constant rate, the intervals between different loads being long enough to fully establish the changed conditions. Under each load the current from the generator was absorbed by a water rheostat, and the load was maintained constantly at the desired point by hand regulation at the switchboard or rheostat terminals. The loads selected (exclusive of direct-current exciter, using from 3 to 5 kw.) were 100, 200, 300, and 400 kw.

TABLE 1 DIMENSIONS OF UNIFLOW ENGINE TESTED

1	Number of cylinders	4
2	Diameter of each cylinder	14 in.
3	Diameter of each piston rod	2¾ in.
4	Stroke of each piston	16 in.
5	Normal speed	300 r.p.m.
6	Piston speed in feet per minute at 300 r.p.m.	800 ft.
7	Clearance volume in per cent of piston displacement as calculated from drawing, first and second series	22 per cent
8	Clearance volume calculated for third and fourth series	17 per cent
9	Inside diameter of steam pipe	4.8 in.
10	Inside diameter of each of the two exhaust pipes	8 in.

9 There were four series of tests, each with the four loads specified, namely:

(*a*) Saturated steam, 125 lb. pressure, atmospheric exhaust
(*b*) Saturated steam, 125 lb. pressure, 5 lb. back pressure
(*c*) Saturated steam, 150 lb. pressure, atmospheric exhaust
(*d*) Superheated steam, 150 lb. pressure, atmospheric exhaust.

The leading data and results are presented in Tables, 2, 3, 4, and 5.

10 The steam on all the saturated-steam tests was practically dry steam, a throttling calorimeter, showing almost constantly a

TABLE 2 TESTS OF UNIFLOW ENGINE—FIRST SERIES

(Saturated steam, 125 lb. pressure, atmospheric exhaust, 22 per cent clearance)

		Proportionate load			
		¼ load	½ load	¾ load	Full load
1	Gage pressure at throttle valve, lb.	125	125	126	125.5
2	Electric load (exclusive of exciter), kw.	100	200	300	400
3	Water per hour, lb.	4360	7144	9808	13,124
4	Indicated horsepower	211.6	350.8	491.8	635.2
5	Water per i.hp-hr., lb.	20.6	20.4	20.1	20.7
6	Water per kw-hr. (exclusive of exciter), lb.	43.6	35.7	33.0	32.8

TABLE 3 TESTS OF UNIFLOW ENGINE—SECOND SERIES

(Saturated steam, 125 lb. pressure, 5 lb. back pressure, 22 per cent clearance)

		Proportionate load			
		¼ load	½ load	¾ load	Full load
1	Gage pressure at throttle valve, lb.	125.3	129	124	124.4
2	Electric load (exclusive of exciter), kw.	100	200	300	400
3	Water per hour, lb.	4655	7776	11,203	15,214
4	Indicated horsepower	215.5	352.7	499.4	652.6
5	Water per i.hp-hr., lb.	21.6	22.0	22.4	23.3
6	Water per kw-hr. (exclusive of exciter), lb.	46.1	38.9	37.3	38

TABLE 4 TESTS OF UNIFLOW ENGINE—THIRD SERIES

(Saturated steam, 150 lb. pressure, atmospheric exhaust, 17 per cent clearance)

		Proportionate load			
		¼ load	½ load	¾ load	Full load
1	Gage pressure at throttle valve, lb.	151	153.2	152.4	150.8
2	Electric load (exclusive of exciter), kw.	100	200	300	400
3	Water per hour, lb.	4314	6834	9668	12,652
4	Indicated horsepower	222.4	361.9	533	634.5
5	Water per i.hp-hr., lb.	19.4	18.9	18.1	19.9
6	Water per kw-hr. (exclusive of exciter), lb.	43.1	34.2	32.2	31.6

TABLE 5 TESTS OF UNIFLOW ENGINE—FOURTH SERIES

(Superheated steam, 150 lb. pressure, atmospheric exhaust, 17 per cent clearance)

		Proportionate load			
		¼ load	½ load	¾ load	Full load
1	Gage pressure at throttle valve, lb.	153	152	152.2	151.4
2	Temperature by throttle thermometer, deg. fahr.	469	443	427	437
3	Superheating at throttle thermometer, deg. fahr.	109	83	67	77
4	Electric load (exclusive of exciter), kw.	100	200	300	400
5	Water per hour, lb.	3411	5807	8430	11,148
6	Indicated horsepower	223.2	365.2	501.5	638.3
7	Water per i.hp-hr., lb.	15.3	15.9	16.8	17.6
8	Water per kw-hr. (exclusive of exciter), lb.	34.1	29.0	28.1	27.9

temperature of 277 deg. fahr. with 125 lb. pressure at the throttle valve, indicating a moisture content of less than one-half of one per cent. No correction was made for this small amount of moisture.

11 The superheating boiler of the plant was not in service while the saturated-steam tests were being conducted.

12 The indicated horsepower was determined by taking diagrams and kilowatt readings simultaneously, and correcting for any departure from the even 100-, 200-, 300-, and 400-kw. loads, which were the points at which the averages were continuously maintained. The diagrams were taken from three-way cocks according to the usual test-block method. The horsepowers should be

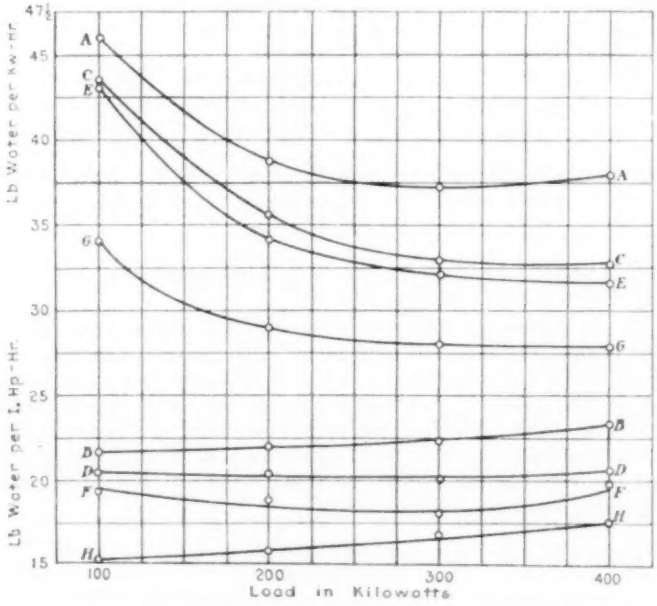


FIG. 2 CURVES SHOWING PERFORMANCES OF THE UNIFLOW ENGINE

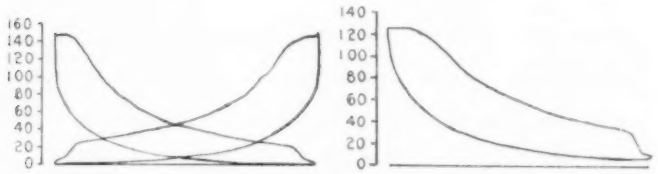
Curves A and B: Saturated steam, 125 lb. pressure, 5 lb. back pressure.
Curves C and D: Saturated steam, 125 lb. pressure, atmospheric exhaust.
Curves E and F: Saturated steam, 150 lb. pressure, atmospheric exhaust.
Curves G and H: Superheated steam, 150 lb. pressure, atmospheric exhaust.

slightly reduced owing to the error of the three-way cocks. This error, however, is practically offset by a corresponding error due to the reducing motion in use which acts the opposite way, and these errors may be disregarded.

13 The economy results of the various tests are shown graphically by points on Fig. 2, in which horizontal distances represent the load in kilowatts and vertical distances pounds of steam consumed per i.hp-hr. and per kw-hr. In each case a curve is drawn showing what may be considered the average curve of performance. It is interesting to note the effect which superheating produced on

the curve H of that series as compared with the corresponding curve F given with saturated steam. In the superheated-steam case the curve is substantially a straight line.

14 The manner in which the steam is distributed in the cylinders by the use of the uniflow principle and its attending valve mechanism and arrangement of cylinders may be seen by reference to the two sample indicator diagrams reproduced in Figs. 3 and 4.



FIGS. 3 AND 4 SAMPLE INDICATOR DIAGRAMS FROM UNIFLOW ENGINE

Fig. 3: Throttle gage, 153 lb.; back pressure, 0 lb.; m.e.p., 45.1 lb.
Fig. 4: Throttle gage, 125 lb.; back pressure, 6.7 lb.; m.e.p., 42.9 lb.

Fig. 3 shows the performance with atmospheric exhaust, and Fig. 4 with a back pressure of 6.7 lb.

15 In connection with the operation of the uniflow principle in this engine, the question of cylinder condensation comes to the front. Table 6 is added with a view of showing the steam accounted for by the indicator diagrams in one of the series of tests with saturated steam, as well as the proportion this bears to the actual steam consumption. The high percentages in the last line of the last two columns (87 and 88 per cent) reveal excellent conditions regarding condensation and leakage.

TABLE 6 STEAM ACCOUNTED FOR BY DIAGRAM IN FIRST SERIES OF TESTS

		Load in kilowatts			
		100	200	300	400
1	Apparent cut-off, per cent	4.1	8.0	11.2	16.9
2	Steam accounted for by diagrams per i.hp-hr., lb.	13.72	16.96	17.49	18.2
3	Actual steam consumed per i.hp-hr., lb.	20.6	20.4	20.1	20.7
4	Percentage accounted for	66.6	83.1	87.0	88.0

16 At the close of the second series of tests the admission valves were tested for leakage under a standing pressure, with engine blocked, the steam which leaked being revealed by escape into the air through the indicator piping. There was practically no leakage. When the throttle valve was shut and the steam was retained in the steam chests, the pressure within dropped at the extremely low rate of 5 lb. per min., requiring ten minutes to be reduced from 120 lb. to 70 lb.

DISCUSSION

JAMES B. STANWOOD.[1] The non-condensing uniflow steam engine is now fully established as a recognized element of direct-connected generating sets. The tests made by the author on one of these sets, driven by a uniflow engine, are, we believe, the first that have been reported to the Society.

These tests are instructive. They show how the large amounts of steam shown by the indicator diagrams form an important feature of the uniflow principle. This makes it possible to employ early cut-offs and large ratios of expansion. In these tests the maximum cut-off was only 17 per cent; at these early cut-offs there are no large volumes of steam thrown away into the exhaust. The flat performance curve obtained is a result of these conditions, and is a desirable feature not possessed by other types of engines. It is obvious that uniflow engines, in order to secure capacity and economy for full load, must necessarily use large cylinders, heavy frames, and generous running gears.

Although so great an improvement has been secured by uniflow cylinders, yet these tests show a still more marked improvement when superheated steam is used. When saturated steam was employed at 125 lb. per sq. in. pressure under a wide range of load, it showed a nearly uniform performance of 20 lb. of steam per i.hp-hr.; when the pressure was raised to 150 lb. there was an improvement in economy of about 10 per cent; and when to the 150 lb. pressure, about 100 deg. fahr. of superheat was added, a further improvement of nearly 15 per cent was secured. The economical advantage of superheat is thus apparent.

A feature of this design of cylinder, as well as that of some other uniflow engines, consists in passing the condensation of the jacket through the cylinder. What would have been the result if this condensate had been eliminated through a trap? The 66 per cent of the steam accounted for by the indicator diagram (shown by Table 6) undoubtedly included part if not all of this jacket condensate. When superheated steam was used, did it not prevent or materially check this jacket condensation?

The superheat as reported in these tests was added by an independent superheater arranged to give 109 deg. fahr. at quarter load and 77 deg. fahr. at full load. In practice, when the superheater is an integral part of a boiler the superheat usually increases as the load increases. This has been the case with locomotives, and if it had been done in these tests the economy at higher loads would have been improved and the flat performance curve would have been more nearly horizontal.

Small installations of direct-connected generator sets have not often been used with superheaters. The writer has found that

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72-in. x 18-ft. horizontal tubular boilers may be equipped with superheaters at a relatively small cost and operate at little or no expense for upkeep. Two 72-in. x 18-ft. boilers rated at 150 hp. are capable of developing 150 per cent of rating by using stokers or high stacks: they will evaporate 13,500 lb. of water into steam at 150 lb. per sq. in. working pressure with superheaters attached. Two of these boilers have ample capacity to operate the generating set upon which the author conducted his experiments.

W. C. Brown.[1] From 1876 to 1912, in which latter year Professor Stumpf first produced the uniflow engine, there had been no improvement in steam engines except in minor details. Notwithstanding the fact that the demand for steam engines had changed from those running at comparatively slow speed for driving mills, lineshafts, etc. to those operating at high speed for direct connection to electric generators, no successful increase in speed was made in the United States until the Ames Iron Works brought out the engine described in the paper, although much more had been learned about inertia forces and the means for overcoming them, and about materials capable of withstanding the higher speeds. Engine builders seemed to consider that it was the cost of the engine only that counted, and that, coupled with their objection to anything that required a change of patterns or standards, has retarded development and the reciprocating-engine business has suffered accordingly. Forcing electric generators to come to existing speeds was playing into the hands of the steam turbine. In the writer's opinion, if the reciprocating engine is to hold its own, it must be further developed along the lines indicated by the engine described in the paper.

The first uniflow engine was conceived from the fact that the turbine had shown the advantages of the steam passing in one direction, permitting the inlet end always to be maintained at a high temperature. Secondly, higher pressures and temperatures had been shown to be practical by the oil engine. Thirdly, multiple cylinders and higher speed were copied from the gasoline engine. The next step was suggested by the steam turbine, namely, of changing the pressure of the steam at the point of release into kinetic energy, and utilizing this kinetic energy to draw steam from another cylinder, so that the back pressure in the cylinder would be lower than the pressure at the point of exhaust.

The multicylinder engine lends itself to these last two advantages, because by multiplying the cylinders it is possible to increase the speed, thereby cutting in two the cost of the generator and greatly reducing the cost of the unit. It also makes possible the utilization of the energy left in the steam at the point of

[1] Directing **Manager**, Stumpf Uniflow Engine Company, Syracuse, N. Y. Mem. A.S.M.E.

release to reduce the back pressure in the cylinder, so that without the use of a condenser a partial vacuum may be obtained with the result of improved economy in operation.

One engine builder in the United States has further utilized the multicylinder engine in a situation where part but not all of the exhaust steam is desired for heating purposes, one or more of the steam cylinders exhausting into the heating system and the others into the condenser.

Referring to the question brought up by Mr. Stanwood as to the use of steam traps for taking the condensate out of the jackets, it may be stated that with superheated steam there is no condensate in the jackets where the steam passes rapidly over them, and with saturated steam there is so little that it does not seem to do any harm to let it go into the cylinders.

The uniflow engine is different from the old type of jacketed engine in that the cylinder head, when once heated, is kept hot. There is no cold exhaust steam to cool it, so that the temperature of the whole cylinder head is practically up to boiler temperature, and any loss in temperature in a cylinder is more than made up by the compression. In Germany Professor Nagel has shown that at the time the valve is opened the temperature of the steam in the cylinder is 200 to 300 deg. fahr. higher than that of the entering steam, so that comparatively little cooling takes place. Since the steam passes rapidly over the surfaces, there is practically no condensation in the jackets.

JAMES TRIBE.[1] The writer is particularly interested in the thermal efficiency of the uniflow engine and is convinced that the steam consumption may be somewhat better than the Corliss simple engine in the field of condensing operation. He is doubtful, however, as to its efficiency when operating non-condensing and against high back pressure. There is a tendency today, and indeed a demand, for exhaust steam of much higher pressure than heretofore used, with a view to utilizing the heat of the exhaust for process work. The writer knows of one case, now in operation, wherein the exhaust pressure is 130 lb. gage.

Now, in uniflow engines, exhaust-steam compression begins at about one-tenth of the stroke, which of course means about nine compressions or a final pressure of about nine times the absolute pressure of the exhaust. Obviously, therefore, if at the end of compression the pressure within the cylinder becomes higher than the throttle pressure, serious damage to the mechanism may result.

In order to hold the compression below the throttle pressure, either the end clearance must be increased or the beginning of compression delayed, and to delay compression to the safety point,

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it has become necessary to install an auxiliary exhaust valve in exactly the same relative position as the exhaust valve in a Corliss cylinder. This valve, together with its ports, introduces additional cooling surfaces (to about the same extent as would be with an increase in clearance) which, being cooled by the low-temperature exhaust steam, increase the percentage of condensation and bring the uniflow engine back again to practically an equivalent of the Corliss method of exhaust. The writer estimates that with 150 lb. throttle pressure and 15 lb. gage back pressure, the steam consumption per unit of power is practically the same as that of a Corliss simple engine operating under the same steam conditions.

C. A. Cahill.[1] The crowning excellence of the uniflow engine is the fact that it is the simplest engine built that can produce the economy that it does produce. The next point of importance is that such engines will run over an extremely wide load range with an almost flat steam-consumption curve. Another point is that they can use steam at any reasonable pressure, say, 300 lb. per sq. in., and at any reasonable temperature. In the matter of high back pressures it is of course necessary to use auxiliary exhaust valves, although but little steam is brought back through these exhaust ports to delay compression, and therefore the engine is under no great disadvantage.

Rudolph Wintzer.[2] Some years ago the company with which the writer is connected installed a large uniflow engine in the Minneapolis district. The question involved was one of both power and heat. During the summer no heat was needed, but in winter the mill needed heat and steam corresponding to about 1300 hp. The engine had auxiliary valves and both sides were condensing or non-condensing. During the summer both sides of the engine operate entirely condensing. During the autumn, when the heating season starts, but little exhaust steam is needed, and the loads are shifted over to the condensing side, the non-condensing side being given but little to do. In this manner there is always available just the amount of exhaust steam that is needed; and never any waste of exhaust steam through the roof.

In order to obtain the proper compression, an engine without an auxiliary exhaust valve requires a certain speed for a certain back pressure and for a certain steam pressure. When conditions change it will be necessary either to provide larger clearance spaces or to lose some efficiency in the engine. If an auxiliary exhaust valve is provided, the engine always can be regulated. While the end

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[2] Chief Engineer, Nordberg Manufacturing Company, Milwaukee, Wis. Mem. A.S.M.E.

of the exhaust is entirely governed by the auxiliary exhaust valve, the compression can be changed while the engine is running.

F. R. Low.[1] It is very impressive that an engine of the kind described by the author is able to replace in a single cylinder the elaborate paraphernalia that we thought to be necessary some years ago with multiple cylinders in order to get rid of cylinder condensation.

It would be interesting to figure out what the effect of that simplification has been upon overall efficiency.

The Author. In presenting the paper at the meeting the author called attention to one point regarding the 150-lb. series with saturated steam, Table 4, which he had noted after the paper had been printed, namely, that the percentages of "steam accounted for by diagram at cut-off" on this series had averaged 12 per cent less than those shown on the 125-lb. series, Table 2, which are given in Table 6 of the paper. As a reason for this reduction, the author suspected that there had been some leakage of valves at the higher pressure which did not exist at the lower pressure of 125 lb. This might readily happen, for the valves were ground in for 125 lb. only, and not for 150 lb. If 10 per cent were taken as the loss in steam accounted for thus caused it seemed probable that, with equal conditions as to leakage, the steam consumption in the 150-lb. series, saturated steam, would have been reduced to the following quantities:

Load, kw.	100	200	300	400
Lb. per i.hp-hr.	16.5	16.5	16.0	17.6

and in the 150-lb. series, superheated steam, to those given below:

Load, kw.	100	200	300	400
Lb. per i.hp-hr.	13.5	14.1	15.1	15.8

It might be well to note also that a comparison of the results of the four tests given in Table 4 revealed a slight inconsistency in the test at 300-kw. load. The indicated horsepower given was obviously somewhat erroneous, being too large, and this made the water per i.hp-hr. too small. A consumption of 19.2 lb. per i.hp-hr., instead of 18.1, was a more consistent figure, and this would likewise change the figure 16 lb. above given to 16.9 lb.

[1] Editor of *Power*, New York, N. Y. Past-President, A.S.M.E.

No. 1969

TORSIONAL VIBRATIONS AND CRITICAL SPEEDS OF SHAFTS

By Arnold Lack[1] and Charles B. Jahnke,[2] Beloit, Wis.

Non-Members

This paper is an analysis of torsional vibrations and critical speeds of shafts. These magnitudes can be calculated, as shown here, for any engine when all necessary data are given. The various characteristics of torsional vibrations are measured with a special instrument, called the "Torsiograph," which shows on "torsiograms" the degree of irregularity of rotation.

The stresses in the shaft due to its vibrations are also computed, and means of avoiding dangerous vibrations are shown. Results of tests are given which were performed on: (a) An apparatus for demonstration of torsional vibrations; (b) an experimental horizontal oil engine; (c) a vertical two-cylinder Diesel engine driving two generators; and (d) a vertical four-cylinder marine engine.

THE importance of torsional vibrations and critical speeds of rotating shafts has only been fully realized in the last few years. Torsional vibrations and their influence on the crankshaft, the whole engine, and the foundation were not very pronounced in steam and gas engines due to the low rotative speeds. More than four cylinders on one shaft were not often employed, consequently the crankshaft was short and less sensitive to the forces tending to produce detrimental torsional vibrations. Steam engines, for example, show a full indicator diagram, and the tangential effort is more constant than in Diesel engines, resulting in lighter flywheels than in Diesel engines of the same power. In multiple-cylinder Diesel engines heavy moving masses act on the crankshaft at each crank, resulting in a higher number of critical speeds. The more critical speeds produced in an engine, the greater is the possibility of synchronism with the engine speed, and the more serious is the influence of heavy rotating masses and long shafts. Marine engines in general have a long shaft extension between the

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[2] Chief Engineer, Fairbanks, Morse & Co.

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engine and the propeller, which is particularly sensitive to the causes of torsional vibrations.

2 The effect of mechanical vibrations of a shaft is to produce not only a noisy-running engine, but also high additional stresses in the shaft and bedplate or frame. Vibrations consume power, and thus the mechanical efficiency is thereby lowered. Torsional vibrations also have a detrimental influence in transmissions employing gears or couplings connected to long shafts. It must be noted that torsional vibrations have a direct influence on the torque, and may cause an increase of irregularity of angular speed. Such variations of speed are inadmissible in electric-light plants, especially for the parallel operation of alternators, and in textile machinery and similar service.

3 Torsional vibrations and critical speeds can be calculated for any engine system when all necessary data are given, as shown in the following study. With the "Torsiograph," a special instrument for this investigation, we can measure the frequency, amplitude, and other characteristics of torsional vibrations. At the same time we obtain the degree of irregularity of rotation by diagrams taken with this instrument; we can also figure the stresses in the shaft due to its vibrations.

General Determination of Harmonic Vibrations in the Shaft

4 Assuming that various twisting moments Tr, T_1r_1, etc., are acting on an elastic shaft, then any line a-a on the surface of the shaft will become the curve a_1-a_2 (Fig. 1). Taking the twisting moment $M = Tr$ at left side of the shaft of constant value in pound-inches,

I_p = polar inertia moment of the constant cross-section of the shaft in inches^4

G = modulus of elasticity for shearing of the shaft material in lb. per sq. in., and

ϑ = relative angle of deviation, due to the torque in the shaft,

then we get, according to the well-known method of determination of torsion in a shaft, the relative torsional deviation for the length L between the sections through T_1 and T, namely,

$$\Delta a = \vartheta r L = \frac{MrL}{I_p G} = \frac{Tr^2 L}{I_p G}$$

or

$$\frac{\Delta a}{L} = \frac{Tr^2}{I_p G} = \vartheta r$$

5 Therefore the relative torsional deflection increases proportional to the length L of the shaft subjected to the torque. This

deflection can be determined graphically as shown in Fig. 1 and in the following examples.

6 Due to the twisting moment $M = Tr$, we get in the shaft of diameter d in inches the specific torsional stress

$$\sigma_{t\,\max} = \frac{\delta G d}{2}$$

7 In this study the twisting moment was assumed to be caused by a static force, but the same torque can also be produced by any dynamic force due to variation of tangential effort of the piston pressures. These tangential forces vary harmonically, and for harmonic fluctuations we can express the vibration at any instant by

$$a \sin(\omega t) = r\alpha \sin(\omega t)$$

where ω = constant angular velocity of a circular motion, from which the harmonic vibration is deduced.

8 The inertia force T necessary for the particular mass m to produce this vibration can be expressed by

$$T = m\omega^2 a \sin(\omega t)$$

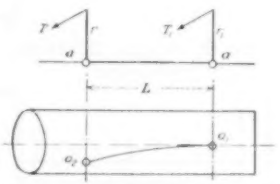


FIG. 1 RELATIVE TORSIONAL DEFLECTION

This expression gives the general relation for a vibration due to harmonic impulses.

FREQUENCY OF TORSIONAL VIBRATIONS FOR ANY ROTATING SYSTEM

9 From the above-developed general relation the final formula for the time for one oscillation in the rotating system, due to harmonic impulses, can be determined according to the theorem of harmonic vibrations.

Let t = time in seconds for one full vibration
F = frequency of complete vibrations per minute
d = diameter of shaft in inches
I_p = polar moment of inertia of the shaft in inches^4
L = free length of shaft in inches
W = weight of wheel or rotating mass in pounds
r = radius of gyration of the wheel in inches
g = acceleration due to gravity = 386 in. per sec. per sec.
G = modulus of rigidity of the shaft material
$= \frac{2}{5} \times 30 \times 10^6 = 12 \times 10^6$ lb. per sq. in. for steel.

Then the time for one vibration in the shaft with one mass of weight W will be (Fig. 2)

$$t = \frac{2\pi}{\sqrt{\frac{I_p G g}{LWr^2}}}$$

And the frequency of vibrations per minute

$$F = \frac{60}{2\pi}\sqrt{\frac{I_p G g}{LWr^2}}$$

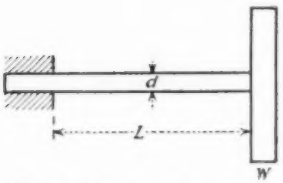


FIG. 2 SINGLE ROTATING WEIGHT

10 For two rotating weights W_1 and W_2 at the distance L, reduced to the same radius r of gyration, we get the frequency of vibrations (Fig. 3)

$$F = \frac{60}{2\pi}\sqrt{\frac{I_p G g (W_1 + W_2)}{L r^2 W_1 W_2}}$$

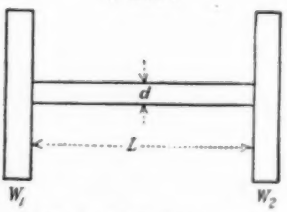


FIG. 3 TWO ROTATING WEIGHTS

With two equal weights $W_1 = W_2 = W$, the frequency is expressed by

$$F = \frac{60}{2\pi}\sqrt{\frac{I_p G g}{\frac{L}{2} W r^2}}$$

11 The expressions for the frequency for three or more rotating masses give such a complicated function that it is preferable to develop the member $\Sigma(LWr^2)$ by a graphical method. The general relation will then be:

$$F = \frac{60}{2\pi}\sqrt{\frac{I_p G g}{\Sigma(LWr^2)}}$$

Assuming steel as shaft material, we get

$$F = \frac{60}{2\pi}\sqrt{\frac{I_p \times 12 \times 10^6 \times 386}{\Sigma(LWr^2)}} = 0.65 \times 10^6 \sqrt{\frac{I_p}{\Sigma(LWr^2)}}$$

From this equation for the number of torsional vibrations of the shaft per minute it will be seen that the frequency increases with the diameter of the shaft, and decreases with a longer shaft and a greater moment of inertia of the rotating masses. The above determined relation for the frequency of torsional vibrations exists for a system with two masses. For a shaft system with n masses the number of natural frequencies is $(n-1)$. In the following examples only the lowest natural frequency is considered. Extend-

ing this preliminary study in a further investigation we shall treat this problem by more detailed examples.

12 Knowing the number of vibrations in the shaft, we can find if a condition of resonance exists between these vibrations and the number of impulses of the engine. In the case of resonance the magnitude of the vibrations can increase to a dangerous value. The engine speed which coincides with the frequency of these vibrations is called *the critical speed*.

13 It must be noted that at the point corresponding to the node of vibrations between the oscillating masses the deviation of angular velocity = 0; but in this part the highest stress due to torsional deflection occurs. Rotating masses—flywheels, rotors, etc.—placed at the node, or near this part of the shaft, have no influence on the torsional vibrations. It must be considered that additional masses add also a new natural frequency, modifying the amplitude of vibrations produced in the original system.

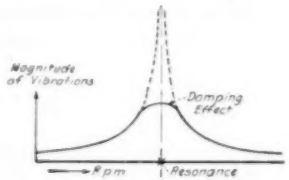


FIG. 4 VIBRATIONS AT POINT OF RESONANCE

14 For multiple-cylinder engines in general the frequency of torsional vibrations has a very high numerical value, namely, a multiple of the r.p.m. of the engine. Therefore multiple-cylinder engines have several points of coincidence between impulses by the engine and vibrations, i.e., several critical speeds.

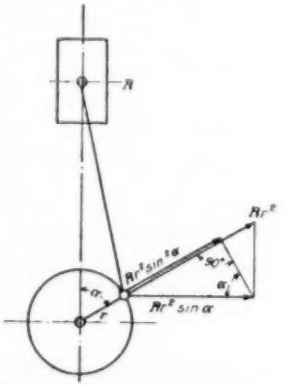


FIG. 5 ROTATING AND RECIPROCATING PARTS

15 The exact investigation of the problem shows that heavy vibrations occur in a certain region near or at the critical speed (Fig. 4). This fact is proven by tests, and it is known that torsional oscillations attain considerable magnitude during a more or less prolonged period near the resonance speed. It will be shown in another part of this paper that various damping forces have a great influence at the point of resonance.

REDUCTION OF THE MOVING MASSES ACTING AT THE CRANKSHAFT

16 The moment of inertia of the rotating parts $\Sigma(Wr^2)$ can be reduced to the crank radius r as radius of gyration. The rotating parts consist of flywheel, rotating load, couplings, crank end of connecting rod, crank, counterweights, etc.

17 The reciprocating parts produce a harmonically varying moment of inertia. These reciprocating parts are: complete piston, piston rod, crosshead, and piston end of connecting rod. The moment of inertia due to these reciprocating parts gives two maximum values per revolution. Reducing these parts of weight R per cylinder to the crank radius r as radius of gyration (Fig. 5), we get for the moment of inertia

$$\Sigma(Rr^2) = Rr^2 \sin^2 \alpha \text{ or}$$

$$\Sigma(Rr^2) = \frac{Rr^2}{2}(1-\cos 2\alpha)$$

18 A part of the moment of inertia, due to the finite length of connecting rod, is of secondary importance in small engines, and equals

$$\Sigma(R_s r^2) = Rr^2 2\lambda \cos\alpha \sin^2\alpha$$

if

$$\lambda = \frac{r}{L} = \text{ratio of connecting rod.}$$

The numerical value of this part is very low and will not be considered in this preliminary study.

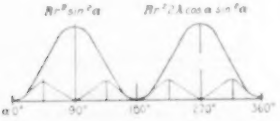


FIG. 6 MOMENT OF INERTIA DUE TO THE RECIPROCATING PARTS OF A SINGLE-CYLINDER ENGINE

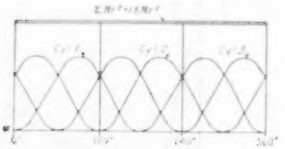


FIG. 7 CASE OF A THREE-CYLINDER ENGINE

19 For a *single-cylinder engine* the moment of inertia of the reciprocating parts R is represented by Fig. 6. It will be seen that the secondary moment due to the connecting rod occurs four times per revolution.

20 *Two-cylinder engines*, with crank setting 0-180 deg., give the following effect of the reciprocating parts:

$$\Sigma(Rr^2) = Rr^2(1-\cos 2\alpha)$$

i.e., twice the value of the single-cylinder engine.

21 *Three-cylinder engines* give, as represented by Fig. 7,

$$\Sigma(Rr^2) = \frac{Rr^2}{2}\left[1-\cos 2\alpha + 1-\cos 2(\alpha+60°) + 1-\cos 2(\alpha+120°)\right] \text{ or } \Sigma(Rr^2) = \frac{Rr^2}{2}\times 3 = 1.5Rr^2$$

i.e., the total effect is of constant numerical value for one revolution.

22 A *four-cylinder engine* is equivalent to 2 two-cylinder engines acting at 90 deg. to each other and gives

$$\Sigma(Rr^2) = Rr^2[1-\cos 2\alpha + 1-\cos 2(\alpha+90°)]$$
$$= 2Rr^2$$

23 A *six-cylinder engine*, being equivalent to 2 three-cylinder engines, gives

$$\Sigma(Rr^2) = 3Rr^2$$

24 It will be shown that for three-, four-, and six-cylinder engines the value of the moment of inertia of the reciprocating parts remains constant. But it must be considered that these parts are acting in different planes, i.e., the reciprocating masses cannot in all cases be assumed concentrated at the same point, in the center of the crankshaft. For an exact investigation these masses of various pistons must be considered separately. For an estimate only, or with relatively small weights, the assumption that all reciprocating parts act in the same plane is admissible.

Reduction of the Crankshaft as an Equivalent Uniform Shaft

25 The cross-section of the crankshaft varies in the different parts; for example, the crankpin usually differs in diameter from the shaft. Of particular importance is the reduction of the crank webs to the equivalent diameter of the shaft.

26 The influence of the fastening of the flywheel, rotor, etc., to the crankshaft must also be taken into consideration. It was shown by experimental tests that the part of the crankshaft rigidly constrained in a hub by a key, shrunk fit, etc. is not subject to torsional deflection. For this reason these parts will be neglected in determining the equivalent length of the crankshaft.

27 For various diameters d_1, d_2, etc. of the shaft, with the respective lengths L_1, L_2, etc., the equivalent length with reference to the standard diameter d is given by the proportion of the inertia moment for these diameters, or equivalent length

$$L = L_1\left(\frac{d}{d_1}\right)^4 + L_2\left(\frac{d}{d_2}\right)^4 + \text{etc.}$$

28 Many more difficulties are encountered when figuring the equivalent length for the crank webs. This problem was the object of a special study of Dr. Geiger, the inventor of the torsiograph bearing his name. His theoretical calculations were checked up by practical tests with various crankshafts. A known torque was applied to the crankshaft, and the equivalent length was determined from the values of the torsional deviations thus obtained.

29 We assume as an average for our Diesel engines that the diameter of the crankshaft (Fig. 8)

$$d = (0.9 \text{ to } 0.95)r$$

Then, according to Geiger's results, we get for the equivalent length

$$L = 1.2L_c$$
$$= 1\tfrac{2}{3}(a+b)$$

when L_c = distance between center lines of main bearings. Cranks with a longer stroke ratio give a longer equivalent length, and inversely. The empirical formula $L = 1.2L_c$ is valid only for the crankshafts figured in the following examples. For another ratio for r more detailed investigation is required. Treatment by various authors gives varying results for the reduced length, and practical tests seem necessary to get a standard treatment of this problem. Tolle,[1] in his newest publication on this subject, comments as follows: "The problem as to how a crankshaft acts during the transmission of torque has as yet not been satisfactorily explained; the most practical way is to determine the amount of torsional deflection due to several moments by experimental tests."

Irregularity of Rotary Motion

30 From the torsional deviation produced in a rotating system, the irregularity of the angular speed of the shaft can be determined as follows: We assume that the oscillations produced in the rotating system by impulses are sinusoidal functions. The corresponding angular speed of the circular motion producing these oscillations is expressed by ω. It must be noted that ω is here not the angular speed of the rotating system. The maximum amplitude of oscillation was determined by

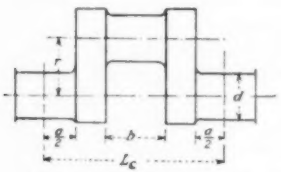


Fig. 8 Crankshaft of Diesel Engine

$$a_1 = \frac{T}{m\omega^2}$$

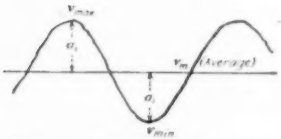


Fig. 9 Velocity Diagram

where T = force producing the twisting moment, i.e., the origin of the oscillations. The maximum velocity of these oscillations can be expressed by

$$v = \frac{T}{m\omega} = a_1\omega$$

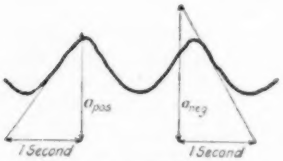


Fig. 10 Torsiogram

Taking the mean peripheral velocity of the shaft $v_m = d\pi N/60$, the irregularity of the angular velocity of the rotating shaft is given by

$$\delta = \frac{v_{max} - v_{min}}{v_m} = \frac{2a_1\omega}{v_m}$$

[1] Max Tolle, Regelung der Kraftmaschinen (third edition), 1921, p. 203.

Therefore the irregularity δ is proportional to the amplitude a of oscillation (Fig. 9). From the values for v_{max} and, v_{min} given by the "torsiogram," i.e., velocity diagram, the amplitude of oscillation is expressed by

$$a_1 = \frac{v_{max} - v_{min}}{2\omega}$$

or

$$a_1 = \frac{\delta v_m}{2\omega}$$

The assumption of a simple harmonic motion, made for this determination, may not be absolutely correct for all fluctuations recorded in Diesel engines. But in general, it is possible to reduce the tachogram of Diesel engines to a sinusoidal function; otherwise this diagram must be analyzed harmonically.

Angular Deviation

31 The numerical value of the angular deviation of a rotating shaft is of particular importance for electric applications. The angular deviation can be deduced from the maximum positive and negative values of the angular speed in the velocity diagram. From the velocities in the torsiogram (Fig. 10), as will be shown later on, we develop the corresponding length of vibrations, and by the maximum and minimum values of a given by this curve the angular deviation can be calculated in degrees as follows:

$$\Delta^\circ = \frac{2a \times 360^\circ}{d\pi}$$

for the diameter d of the shaft.

Damping Forces

32 So far only harmonically acting piston forces and the inertia forces of the moving parts have been considered. In addition to these internal forces, other external forces occur, namely, the damping forces. Damping forces can be enumerated as follows:

Friction in the crankshaft structure and in the bearings, due to weight and piston forces

Friction in bearings and pistons due to vibrations produced by moving parts

Air resistance in flywheels, rotors, and other rotating parts

Water resistance of propellers, impellers in turbines and centrifugal pumps, etc.

Magnetic resistance of electric generators.

These damping forces diminish the magnitude of vibrations in the engine shaft. Due to the damping effect the maximum vibrations do not occur exactly at the point of resonance with the critical speed, but generally at a lower speed.

The Torsiograph

33 For practical investigations of torsional vibrations in machinery, the torsiograph can be conveniently employed. This instrument is in principle similar to the seismograph used for recording of earthquake effects. The torsiograph has a pulley connected to the shaft under test by means of an inelastic belt. The lack of elasticity of this belt is of great importance; elastic deformations and fluctuations in the belt would modify the results, given by the torsiograph. Inside the instrument a heavy rotating mass, acting as a flywheel, is mounted on ball bearings and is connected with the pulley through helicoidal springs. The weight of the rotating flywheel mass and the springs are so proportioned that this flywheel in the instrument will tend to rotate at a uniform speed notwithstanding the vibrations of the pulley. The critical speed of the instrument is of such low value, that for the lowest practical speeds the flywheel in the torsiograph is not influenced by the vibrations of the instrument.

34 The relative motion between the torsiograph flywheel and pulley is communicated through levers to a recording pen, which draws a curve on a paper strip wound upon a drum operated by clockwork mechanism or through a gear from the pulley in such a manner that the paper feed is uniform. The determination of the frequency of vibration is accomplished by an electric time recorder. Another type of recorder for the same purpose can be operated by hand.

Measurement of Torsional Vibrations

35 The torsiograph is driven by a special belt from the shaft of the engine to be tested. For the belt connection it is important to choose that part of the engine at which the highest vibrations occur, namely, near the flywheel or near the heaviest rotating masses, sometimes at the end of the shaft.

36 Diagrams should not be taken at the standard engine speed only. Because high variation of frequency and intensity of vibrations are possible in a small range of speed, it is worth while to examine the engine at various speeds. By varying the scale of the instrument any amplitude of waves can be obtained.

37 The critical speed at the resonance between the impulses in the engine and the frequency of vibrations in the engine or foundation can be determined by examining the characteristics of the torsiograms recorded at various speeds.

38 Besides torsional vibrations of the engine shaft, there can occur vibrations of the engine as a whole. Sometimes vibration in the bedplate or foundation is due to outside sources and not to the engine itself. For detecting these vibrations with the torsiograph it is only necessary to disconnect the belt pulley from the

instrument so that this pulley is stationary. Then the paper drum alone is driven by the clockwork and the diagrams recorded show vibrations of the whole engine or the foundation, upon which the torsiograph is mounted.

Measurement of the Irregularitiy of an Engine Shaft

39 By the torsiograms taken with the torsiograph, either the usual irregularity, i.e., fluctuation of speed, can be determined, or the angular deviation of the shaft. For some purposes it is advantageous to know the angular deviation, especially for electric drive and for textile machinery, paper-making machines, etc.

40 Knowing the degree of angular deviation of speed at different parts of the engine, we can at the same time also determine the stress due to torque produced in the shaft. The usual irregularity is given by the expression

$$\delta = \frac{v_{max} - v_{min}}{v_m}$$

where v_m = mean peripheral speed of the shaft.

41 The respective maximum and minimum values of the peripheral speed of crankshaft, i.e., v_{max} and v_{min}, occur at those parts in the torsiogram where the curve shows the highest positive or negative values. The surplus above the average speed will be referred to as a_{pos} and the highest value below the average as a_{neg}, when the irregularity may be expressed as

$$\delta = \frac{a_{pos} + a_{neg}}{v_m}$$

The value a_{pos} represents the maximum lead per time unit, or per second, and a_{neg} the maximum lag per time unit. Therefore to get the degree of irregularity, it is only necessary to find the steepest tangents per second from the torsiogram, such as shown in Fig. 10 and in the following examples.

Effect of Vibrations in Crankshaft, Engine, and Foundation, and Means of Avoiding Such Vibrations

42 It has been known for a number of years that torsional vibration is the origin of most crankshaft failures. The stress calculations for such broken shafts show that relatively low stresses exist due to the static loads. Dynamic loads are more serious, due to impact, vibrations, etc., and of these influences torsional vibrations are of extreme importance to the safety of a crankshaft due to their high frequency. The repetition of elastic deformations of the shaft in opposite directions may finally cause a fatigue of the material, and the limit of resilience may be exceeded. The great number of vibrations in crankshafts due to critical speeds will be seen in the following examples.

43 In all cases these vibrations cause additional stresses in the crankshaft; the weakest parts, the crank webs, receive a great amount of increase of stress, and thus frequent breakages of cranks may be explained.

44 In stationary Diesel engines the heavy moving parts and heavy flywheels tend to produce torsional vibrations. This is especially so in multi-cylinder engines. Besides the danger of excessive stresses in the crankshaft, noisy operation results and vibrations are transmitted to foundation and building. The consumption of power is considerable in the case of heavy vibrations due to rapid and numerous changes of torque. Lower mechanical efficiency results because energy is required to produce the vibrations.

45 In engines supported upon springs, such as for locomotives, cars or other vehicles, the presence of stronger vibrations is not permissible.

46 Especially for long shafts with heavy rotating parts, such as in ships, the influence of torsional vibrations must be the object of careful investigation. The hull of a ship is an elastic structure and has a certain frequency of vibration of its own. At synchronism between these vibrations and the frequency of impulses of the engine, a serious amplification of vibrations occurs.

47 Long transmission shafts can also be subjected to the above-mentioned influences. Here heavy vibrations are disagreeable for gear drives, and an excessive wear in gears and noisy operation may result.

48 Torsional vibrations at critical speeds can influence the regularity of rotation to a high degree. For light plants and for parallel operation of alternators the irregularity of rotation is limited by the admissible excess of angular deviation of approximately ± 2.5 electric degrees. The values of electric degrees equal the usual mechanical deviation in degrees multiplied by the number of pairs of poles of the alternator.

49 Paper and textile machinery demand a high degree of speed regularity. For these machines the excess of angular deviation is also limited by a small margin; sudden strong fluctuation of angular speed directly affects the quality of the product.

50 The principal origin of the torsional vibrations of crankshafts can be attributed to torque variations due to inertia of moving parts and to piston pressures. We cannot modify the piston pressures, but modifications of the rotating parts and their distribution on the engine shaft are possible. For example, flywheels on the free end of the engine produce a certain detrimental torque in the shaft; auxiliary flywheels on this end should be avoided. Flywheels or other rotating masses placed in the node of vibrations are without influence on these vibrations themselves as far as the original system is concerned. However, on adding flywheels or other masses to this system in the nodes, the

influence of the new frequencies of higher order must be taken into consideration. But gears, couplings, or similar connections placed in the nodes receive the highest deflection and are exposed to dangerous vibrations and high wear. This method of driving the camshaft and accessories from the flywheel end of an engine should be avoided, because at this part the heaviest vibrations occur.

51 In theoretical analysis we possess all means to predict torsional vibrations, their amplitude and frequency, and therefore the possibility of critical speeds. But not in all cases can all necessary characteristics of an elastic structure be determined in order to obtain exact results. It cannot be claimed that a method for calculating the critical speeds of torsional vibrations

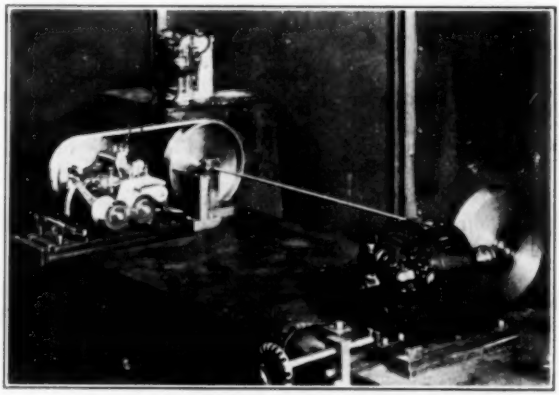


FIG. 11 APPARATUS FOR DEMONSTRATION OF TORSIONAL VIBRATIONS

will in itself remedy all troubles in existing engines; it is also impossible to give any general rule for eliminating these vibrations. An analysis of an engine may, however, give such information upon its behavior as to enable us to correct its action intelligently.

52 With the torsiograph we can check up our mathematical deductions, and it is thus a convenient means for proving our conclusions. The torsiograph is the first instrument to make a visible demonstration of torsional vibrations possible. It is recommended that indicator cards be taken to examine the performance of the engine, while torsiograms should be taken on shafts or other rotating parts to examine more thoroughly the behavior of the engine and to discover dangerous elastic deformations of the shaft, or critical speeds, and other detrimental phenomena.

Results of Tests

53 (*a*) *Apparatus for Demonstration of Torsional Vibrations.* Two equal metal disks 8 in. in diameter, weighing 8.55 lb. each, are mounted at each end of a $\frac{5}{16}$-in.-diameter steel shaft (Fig. 11). One disk is driven by a small friction wheel mounted on the shaft of an electric motor. The other disk is belted to the torsiograph, and has a pin which engages a flat spring at each revolution to produce a torsional impulse in the rotating system. This test arrangement was mounted upon a very heavy casting block to reduce the influence of vibrations of this base to a minimum.

54 The frequency of torsional vibrations with two equal masses was determined by

$$F = \frac{60}{2\pi}\sqrt{\frac{I_p G g}{\frac{L}{2} W r^2}}$$

where I_p = polar inertia moment $= \frac{\pi d^4}{32} = \frac{\pi 5^4}{32 \times 16^4} = 0.00094$ in.^{4}

G = shear modulus of elasticity $= 12 \times 10^6$ lb. per sq. in.

g = acceleration due to gravity = 386 in. per sec. per sec.

L = free length of shaft between the two disks $= 33\frac{1}{4}$ in.

Wr^2 = moment of inertia of each rotating mass = 92 lb-in.^{2}

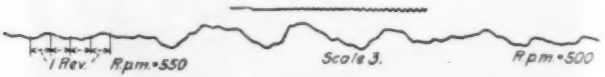


Fig. 12 Torsiogram of Experimental Apparatus

Then the frequency equals

$$F = \frac{60}{2\pi}\sqrt{\frac{0.00094 \times 12 \times 10^6 \times 386}{\frac{33\frac{1}{4}}{2} \times 92}} = 508$$

55 Another form of critical speed, produced by radial deflection due to the weight of the shaft itself, is given by the equation

$$N_c = 10.61674 \times 10^6 \times \frac{d}{L_c^2}$$

where L_c = distance of bearings = 30 in., or

$$N_c = 10.61674 \times 10^6 \times \frac{5}{16 \times 30^2} = 3680$$

This critical speed is so much higher than that due to torsional vibrations that one cannot be influenced by the other.

56 The torsiogram, Fig. 12, recorded by the torsiograph shows the beginning of the critical speed when the speed exceeds 508 r.p.m. The heaviest vibrations continue during approximately

30 revolutions increase. The range of maximum vibrations in this case was rather long because the paper was fed at high rate, which did not permit a quieting of the vibrations in a short interval of time.

57 The maximum fluctuation is given by the torsiogram as $\frac{3}{4}$ in. The scale of the torsiogram is three times the actual size; the diameter of the torsiograph pulley was 5.83 in.; therefore the fluctuation of velocity at the periphery of the disks was

$$a = \frac{\frac{3}{4}}{3} = 0.25 \text{ in.}$$

58 (*b*) *Experimental Horizontal Oil Engine. Single Cylinder* (25 b.hp., $10\frac{1}{2}$ in. diameter, 13 in. stroke, r.p.m. 325). This engine

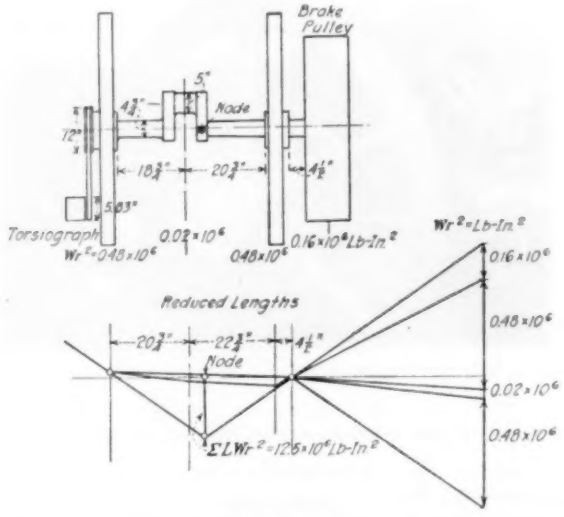


FIG. 13 HORIZONTAL ENGINE WITH BRAKE PULLEY. DIAGRAM OF MOMENTS OF INERTIA

was tested under full load, a Prony brake at one end of the crankshaft supplying the load. With the additional weight of this brake pulley it was possible to produce the condition of a serious critical speed.

59 The irregularity of the rotary motion, calculated by indicator card and moment of inertia of moving parts, was as follows:

$$\delta = \frac{\Delta F}{F} \times \frac{\text{i.hp.} \times 3.9 \times 10^8}{\Sigma(WD^2) \times N^3}$$

where $\frac{\Delta F}{F}$ = ratio of the maximum surplus work to the average work per revolution; for full load this ratio = 1.2

i.hp. = 38; N = 325

ΣWD^2 = moment of inertia of the moving parts in lb-ft.^{2}, namely:

Moving parts in engine	WD^2 =	700 lb-ft.^{2}
Two flywheels 60 in.$\times 3\frac{1}{2}$ in. = 2×840 lb.	WD^2 =	26700 lb-ft.^{2}
Total	WD^2 =	27400 lb-ft.^{2}
With brake pulley 48 in.$\times 12$ in. = 300 lb.	WD^2 =	4400 lb-ft.^{2}
Total	WD^2 =	31800 lb-ft.^{2}

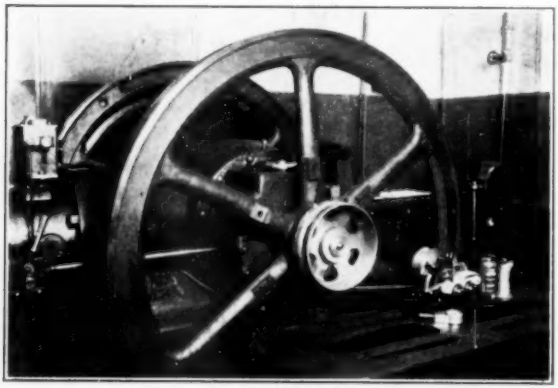


FIG. 14 TORSIOGRAPH ON HORIZONTAL ENGINE

The irregularity for regular service without brake pulley

$$\delta_1 = \frac{1.2\times 38\times 3.9\times 10^8}{27400\times 325^3} = \frac{1}{53}$$

and with brake pulley

$$\delta_2 = \frac{1.2\times 38\times 3.9\times 10^8}{31800\times 325^3} = \frac{1}{61}$$

60 *Frequency of Torsional Vibrations.* Reduction of the crankshaft to diameter $4\frac{3}{4}$ in.:

Pin $L_r = 5\frac{3}{4}\times\left(\frac{4\frac{3}{4}}{5}\right)^4 = 5\frac{3}{4}\times 0.8 = 4\frac{3}{4}$ in. i.e., decrease of length = 1 in.

Crank = $23\frac{1}{2}\times 1.2 = 28\frac{1}{2}$ in., i.e., increase of length = 5 in.

Total equivalent length $L_r = 39\frac{1}{2} - 1 + 5 + 4\frac{1}{2} = 48$ in.

Frequency of torsional vibrations of the *engine alone, without brake pulley:*

$\Sigma(LWr^2) = 10.5 \times 10^6$ lb-in.^{2}. This value can be determined graphically by means of inertia diagram, see Fig. 13.

I_p for $4\frac{3}{4}$ in. diameter = 50 in.^{4}

$$\text{Frequency } F = \frac{60}{2\pi}\sqrt{\frac{50 \times 12 \times 10^6 \times 386}{10.5 \times 10^6}} = 1420$$

Critical speeds:	Order	4	5	6
Critical speeds:	N_c =	355	284	236

No synchronism of critical speed with the regular r.p.m. occurs without brake pulley.

Frequency of torsional vibrations *with brake pulley* (Fig. 13):

$\Sigma(LWr^2) = 12.5 \times 10^6$ lb-in.^{2}

$$\text{therefore} \qquad F = \frac{60}{2\pi}\sqrt{\frac{50 \times 12 \times 10^6 \times 386}{12.5 \times 10^6}} = 1300$$

Critical speeds:	Order	4	5	6
Critical speeds:	N_c =	325	260	216

One critical speed coincides with the regular engine speed.

FIG. 15 TORSIOGRAM OF HORIZONTAL ENGINE

61 The torsiograph was driven by a pulley at the crankshaft end opposite brake pulley (Fig. 14):

Diameter of pulley at crankshaft, 12 in.
Diameter of pulley at torsiograph, 5.83 in.
Scale of the torsiograms = 3.

62 *Critical Speeds* were recorded by the torsiograph at r.p.m. 310, 250, and approximately 210. These speeds are a little lower than calculated, which is due to the damping effect of the brake.

63 In the torsiogram Fig. 15 the maximum deviation of peripheral speed is given by the steepest tangents with the following values: $a_1 + a_2 = \dfrac{1.55+1}{0.3} = 8.5$ in., and reduced to the pulley at the crankshaft

$$a = a_1 + a_2 = \frac{8.5}{3} = 2.83 \text{ in.}$$

Average of peripheral velocity of the pulley:

$$v_m = \frac{d\pi N}{60} = \frac{12 \times \pi \times 325}{60} = 203 \text{ in. per sec.}$$

Therefore the coefficient of irregularity

$$\delta = \frac{a}{v_m} = \frac{2.83}{203} = \frac{1}{72}$$

This coefficient of irregularity is much higher than calculated, due to the amplification of the natural flywheel irregularity caused by torsional vibrations. The angular deviation in degrees has the following value:

$$\Delta^\circ = \frac{a \times 360}{d \times \pi} = \frac{2.83 \times 360}{12 \times \pi} = 27 \text{ deg. total per sec.}$$

64 *Stress in the Crankshaft Due to Torsional Vibrations.* The maximum amplitude of vibrations cannot be read correctly in the graph because the stroke of the recording lever was limited at critical speed. The limited amplitude in the diagram is $1\frac{1}{2}$ in., and reduced to a diameter $d = 1$ in., this amplitude would be in both directions

$$a = \frac{1.5}{2 \times 3 \times 12} = 0.021 \text{ in.}$$

Number of highest vibrations $N = 310$ per minute; therefore angular velocity of these vibrations

$$\omega = \frac{\pi \times N}{30} = \frac{3.14 \times 310}{30} = 32.7 \text{ per sec.}$$

and

$$\omega^2 = 1070 \text{ per sec.}^2$$

Moment of inertia of the flywheel $WD^2 = 1.92 \times 10^6$ lb-in.^{2}, and the dynamic mass reduced to 1 in. diameter is

$$m = \frac{WD^2}{g} = \frac{1.92 \times 10^6}{386} = 5000 \text{ lb-sec.}^2 \text{ in.}$$

The corresponding mass force becomes

$$T = m\omega^2 a = 5000 \times 1070 \times 0.021 = 113000 \text{ lb.}$$

acting at the radius $r = 0.5$ in., therefore the resulting turning moment

$$M_t = T \times 0.5 = 113000 \times 0.5 = 56500 \text{ lb-in.}$$

This turning moment due to vibrations is acting at the end of the shaft opposite the brake. The additional stress due to vibration in the crankshaft equals

$$\sigma_t = \frac{M_t}{S_t}$$

where S_t = section modulus for torsion.

$$S_t = \frac{\pi \times d^3}{16} = \frac{\pi \times 4\frac{3}{4}^3}{16} = 21 \text{ in.}^3$$

$$\sigma_t = \frac{56500}{21} = 2700 \text{ lb. per sq. in.}$$

65 *Additional Stress in Crank Produced by Torsional Vibrations.* Tangential force due to vibrations, reduced to crank radius $r = 6\frac{1}{2}$ in.:

$$T = \frac{56500}{6\frac{1}{2}} = 8700 \text{ lb.}$$

The maximum stress in the crank webs due to this force was found by calculation to be

$$\sigma = 7100 \text{ lb. per. sq. in.}$$

66 Assuming that the same torsional vibrations occur in the crankshaft half at the brake side, then the above-figured stress should be added to the stress due to the acting torque of the piston pressures. The tangential effort due to the piston pressure at the crankpin is 9500 lb. and the corresponding stress in the

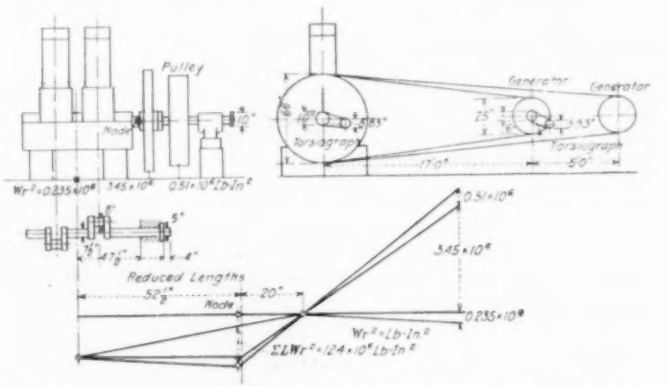


FIG. 16 VERTICAL TWO-CYLINDER DIESEL ENGINE. DIAGRAM OF MOMENTS OF INERTIA

crank webs 7750 lb. per sq. in. Therefore the total stress in the crank webs equals

$$\sigma_{tot} = 7100 + 7750 = 14850 \text{ lb. per sq. in.}$$

Thus the total stress is approximately double the stress due to the acting torque. But, as mentioned before, the highest vibration could not be recorded with the particular scale and torsiograph pulley employed in the experiment. The maximum stress is therefore really higher than the figure calculated above.

67 (*c*) *Vertical Two-Cylinder Diesel Engine* (100 b.hp., 14 in. diameter, 17 in. stroke, r.p.m. = 270). This installation is a temporary one in the power house of Fairbanks, Morse & Co., Beloit Works. The engine is mounted on a temporary foundation consisting of cast-iron blocks, and for this reason this engine has vibrations.

68 Of greatest interest here is the comparison of the irregularity of the Diesel engine with that of the two generators. The generators are driven by two belts from the same pulley at the crankshaft of the engine (Fig. 16).

69 *Irregularity* of the Diesel engine, and moment of inertia of the moving parts:

Ratio of the maximum surplus work per cycle $\frac{\Delta F}{F} = 0.30$ for full load and 0.45 for half-load.

Moment of inertia of the moving parts of the Diesel engine:

Flywheel 78 in. x 6¾ in. = 3600 lb.	WD^2 =	95800 lb-ft.²
Pulley 66 in. x 15 in. = 700 lb.	WD^2 =	14200 lb-ft.²
Moving parts in engine	WD^2 =	6000 lb-ft.²
Total	WD^2 =	116000 lb-ft.²

Irregularity at full load:

$$\delta = \frac{0.30 \times 150 \times 3.9 \times 10^8}{116000 \times 270^2} = \frac{1}{130}$$

and at half-load with r.p.m. = 275;

$$\delta_1 = \frac{0.45 \times 100 \times 3.9 \times 10^8}{116000 \times 275^2} = \frac{1}{142}$$

70 *Frequency of Torsional Vibrations.* Reduction of the crankshaft to diameter 7½ in.:

One pin $L_r = 7\frac{1}{2}\left(\frac{7\frac{1}{2}}{8}\right)^4 = 7\frac{1}{2} \times 0.8 = 6$ in.

(i.e., decrease of length = 1½ in.)

One crank = $33 \times 1.2 = 39\frac{1}{2}$ in.

(i.e., increase of length = 6½ in.)

Shaft extension $L_r = 4\left(\frac{7\frac{1}{2}}{5}\right)^4 = 4 \times 5 = 20$

(i.e., increase of length = 16 in.)

(Total increase of length = 21 in.)

Total equivalent length of crankshaft $L_r = 52\frac{1}{2} + 20 = 72\frac{1}{2}$ in.

Moment of inertia for each side from the node (Fig. 16):

$\Sigma(LWr^2) = 12.4 \times 10^6$ lb. in.³

I_p for 7½ in. diameter = 311 in.⁴

therefore $$F = \frac{60}{2\pi}\sqrt{\frac{311 \times 12 \times 10^6 \times 386}{12.4 \times 10^6}} = 3250$$

Vibrations per revolution $= \frac{3250}{270} = 12.$

71 *Tests Recorded with the Torsiograph* (Figs. 16-20):

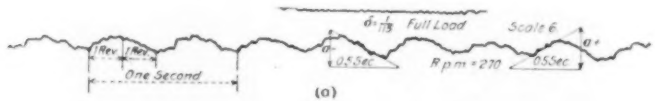
Diameter of pulley at crankshaft	10	in.
Diameter of pulley at generator	6	in.
Diameter of pulley at torsiograph	5.83	in.

Scale of torsiograms = 6.

72 *Irregularity Recorded by Torsiograms at Engine* (Figs. 17-18):

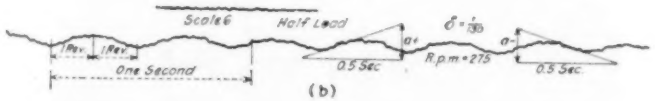
Peripheral velocity of pulley $v_m = \frac{10 \times 270 \times \pi}{60} = 141$ in. at full load.

Full load: (Fig. 17) $a = \frac{1+1.13}{0.5 \times 6} = 0.71$ in. per sec.



(a)

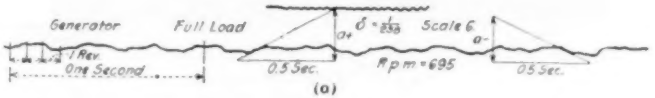
FIG. 17



(b)

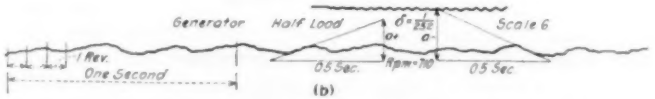
FIG. 18

FIGS. 17 AND 18 TORSIOGRAMS OF DIESEL ENGINE: (*a*) FULL LOAD; (*b*) HALF-LOAD



(a)

FIG. 19



(b)

FIG. 20

FIGS. 19 AND 20 TORSIOGRAMS OF GENERATOR: (*a*) FULL LOAD; (*b*) HALF-LOAD

$$\text{Irregularity } \delta = \frac{0.71}{141} = \frac{1}{198}$$

Half-load: (Fig. 18) v_m for r.p.m. = 275 = 144 in. per sec.

$$\text{Deviation } a = \frac{1+0.95}{0.5 \times 6} = 0.65 \text{ in. per sec.}$$

$$\text{Irregularity } \delta_1 = \frac{0.65}{144} = \frac{1}{222}$$

73 *Irregularity Recorded by Torsiograms at Generator* (Figs. 19-20):

$$v_m = \frac{6 \times 695 \times \pi}{60} = 219 \text{ in. per sec. at full load}$$

Full load: (Fig. 19) $a = \dfrac{1.15+1.5}{0.5 \times 6} = 0.88$ in. per sec.

$$\text{Irregularity } \delta = \frac{0.88}{219} = \frac{1}{247}$$

Half-load: (Fig. 20) v_m = for r.p.m. = 710 = 223 in. per sec.

$$\text{Deviation } a = \frac{1.45+1.115}{0.5 \times 6} = 0.855 \text{ in. per sec.}$$

$$\text{Irregularity } \delta_1 = \frac{0.855}{223} = \frac{1}{260}$$

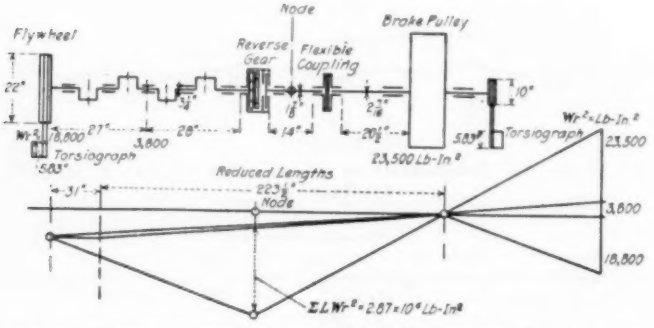


FIG. 21 VERTICAL FOUR-CYLINDER MARINE ENGINE. DIAGRAM OF MOMENTS OF INERTIA

74 Angular deviation in degrees at generator:

$$\text{Full load: } \Delta^\circ = \frac{a \times 360}{d \times \pi} = \frac{0.88 \times 360}{6 \times 3.14} = 17 \text{ deg. total per sec.}$$

$$\text{Half-load: } \Delta^\circ = \frac{0.855 \times 360}{6 \times 3.14} = 16.5 \text{ deg. total per sec.}$$

75 The results of tests in comparison with the calculation for the coefficient of irregularity are given by the following table:

	Irregularity	(a) Full load	(b) Half load
Diesel engine	Calculated	1/130	1/142
Diesel engine	Reading	1/198	1/222
Generator	Reading	1/247	1/260

76 Influence of the moment of inertia of the rotating weights on the generator shaft:

One rotor approximates 1400 lb.	WD^2 approximates 1400 lb-ft.^{2}
One pulley approximates 200 lb.	WD^2 approximates 600 lb-ft.^{2}
At each generator shaft	WD^2 approximates 2000 lb-ft.^{2}
For both generators total	WD^2 approximates 4000 lb-ft.^{2}

Reduced to the speed of the Diesel engine this rotating effect will be:

$$4000 \left(\frac{695}{270}\right)^3 = 68000 \text{ lb-ft.}^2$$

i.e., about 60 per cent of the total moment of inertia of all moving parts in the Diesel engine; but the regularity of the generator is

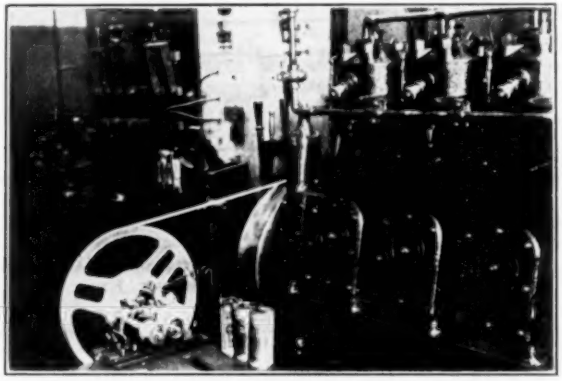


FIG. 22 TORSIOGRAPH APPLIED TO 30-B.HP. MARINE ENGINE

about 20 per cent better than in the Diesel engine, therefore it must be concluded that the difference between both values is due to the belts.

77 (*d*) *Vertical Four-Cylinder Marine Engine* (30 b.hp., 5½ in. diameter, 6½ in. stroke, r.p.m. = 650). (Figs. 21-24.) The crankshaft of this oil engine is connected to a shaft extension by a flexible coupling with leather disks. A brake pulley is mounted at the end of this shaft; a reverse gear with 8 gears is located between crankshaft and shaft extension. The clearances in these gears and the coupling are of influence to the frequency of vibrations, as will be seen in the following tests.

78 *Irregularity* of the engine, figured by indicator card and moment of inertia of moving parts:

Flywheel 22 in. x 4 in. = 245 lb.	WD^2 =	520 lb-ft.^{2}
Moving parts in the engine	WD^2 =	100 lb-ft.^{2}
Engine total	WD^2 =	620 lb-ft.^{2}
Brake pulley 36 in. x 12 in. = 320 lb.	WD^2 =	680 lb-ft.^{2}
Engine and pulley total	WD^2 =	1300 lb-ft.^{2}

Ratio of the maximum surplus work per cycle:

$$\frac{\Delta F}{F} = 0.08$$

Irregularity of the engine alone:

$$\delta = \frac{0.08 \times 46 \times 3.9 \times 10^8}{620 \times 650^2} = \frac{1}{118}$$

Irregularity with shaft extension and brake pulley:

$$\delta_1 = \frac{0.08 \times 46 \times 3.9 \times 10^8}{1300 \times 650^2} = \frac{1}{247}$$

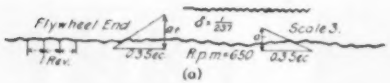


FIG. 23

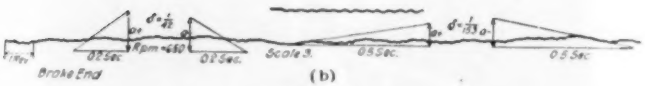


FIG. 24

FIGS. 23 AND 24 TORSIOGRAMS OF FOUR-CYLINDER MARINE ENGINE: (*a*) FLYWHEEL END; (*b*) BRAKE END

79 *Frequency of Torsional Vibrations.* Moving parts in the engine, acting at the four cranks:

$$\Sigma W r^2 = 3800 \text{ lb-in.}^2$$

Reduced lengths of shafts for standard diameter $3\frac{1}{4}$ in.:

Crankshaft: $55 + 4 \times 0.2 \times 10\frac{3}{4} = 55 + 8\frac{1}{2}$ in. $= 63\frac{1}{2}$ in.

Reverse shaft: $14\left(\frac{3\frac{1}{4}}{1\frac{7}{8}}\right)^4 = 14 \times 9 = 126$ in.

Extension shaft: $20\frac{1}{2}\left(\frac{3\frac{1}{4}}{2\frac{7}{16}}\right)^4 = 20\frac{1}{2} \times 3.17 = 65$ in.

Reduced lengths total $254\frac{1}{2}$ in.
I_p for $3\frac{1}{4}$ in. diameter $= 11$ in.^{4}

Engine alone with flywheel: $\Sigma(LWr^2) = 98000$ lb-in.^{2}

$$F = \frac{60}{2\pi}\sqrt{\frac{11 \times 12 \times 10^6 \times 386}{98000}} = 6900$$

Critical speeds: Order		16	12	10
Critical speeds:	N_c =	430	570	690

Engine with shaft extension and brake pulley (Fig. 21):

$$\Sigma(LWr^2) = 2{,}870{,}000 \text{ lb-in.}^3$$

$$F = \frac{60}{2\pi}\sqrt{\frac{11\times12\times10^6\times386}{2{,}870{,}000}} = 1310$$

Critical speeds: Order	2	2.66	4
Critical speeds: N_c =	655	490	327

80 *Tests Recorded with the Torsiograph* (Fig. 22).

Diameter of pulley (flywheel)	22	in.
Diameter of pulley at brake end	10	in.
Diameter of pulley at torsiograph	5.83	in.

Scale of the torsiograms = 3

81 *Irregularity at Flywheel End* (Fig. 23):

$$v_m = \frac{22}{60}\times\pi\times650 = 745 \text{ in. per sec.}$$

$$\delta = \frac{1}{890}$$

82 *Irregularity at Brake End* (Fig. 24):

$$\delta_1 = \frac{1}{262}$$

Considering the small fluctuations in the tachogram, the irregularity at brake end is increased to

$$\delta_2 = \frac{1}{72}$$

These results show that the regularity at brake end is much lower than the calculated theoretical value; this difference must be due to the flexible coupling and to the clearance in the reverse gear. For the same reason the frequency of torsional vibrations also shows no agreement with the value given by calculation. The diagrams recorded at flywheel end show that this diameter for the instrument drive was too large. Our subsequent inquiry to the inventor of the torsiograph confirmed the fact that small diameters for the instrument drive are to be preferred.

DISCUSSION

H. SCHRECK.[1] The calculation of torsional vibrations and critical speeds of oil engines by Messrs. Lack and Jahnke is very timely, and these gentlemen should be complimented on their contribution of this work to the engineering world.

In Par. 29 it is stated that according to Geiger, the reduced length of crankshaft is expressed by $L = 1.2L_c$. There is great

[1] In charge of design, Combustion Utilities Corporation, New York, N. Y. Mem. A.S.M.E.

doubt in the writer's mind whether it is justifiable to use an empiric formula, since the moment of inertia of the crank web and stroke of engine vary considerably in one case from another. The writer has been using successfully the formula by Holzer, which reads as follows:

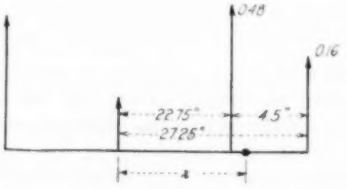
$$L = 2R\,\frac{G}{E} \times \frac{J_0}{J_s} + Z\,\frac{J_0}{J_z}$$

in which R = crank radius
G = modulus of rigidity
E = modulus of elasticity
J_0 = polar moment of inertia of crankshaft
J_z = polar moment of inertia of crankpin
J_s = moment of inertia of crank web = $\dfrac{bh^3}{12}$
Z = length of crankpin.

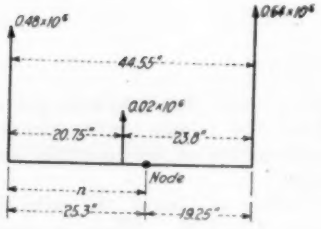
The reduction of length is the most important work, since minor changes in the length of shafting make a large difference in the result.

The writer prefers to determine the node analytically and not graphically as in the paper. For comparison, the case presented in Par. 59 may be worked out in the following manner.

A system of four masses is best reduced to a system of three masses as follows:



$$(0.48+0.16)X = 0.48\times 22.75 + 0.16\times 27.25$$
$$X = 23.8 \text{ in.}$$



$$n = \frac{0.02\times 20.75 + 0.64\times 44.55}{0.48+0.02+0.64}$$
$$n = 25.3 \text{ in.}$$

Therefore, the moment of inertia for each side from the node is

$$\Sigma LWR^2 = 19.25\times 0.64\times 10^6$$
$$= 12.3\times 10^6$$

The authors state that for the single-cylinder engine the calculated critical speed and the recorded critical speed agree fairly

well, but they might agree still better if the above formula for the reduction of length of crankshaft were used.

In the examples of the two-cylinder and four-cylinder engines, the authors do not state the recorded critical speed, and it would be of much interest to learn how closely the calculated and recorded critical speeds agreed; the writer's experience has been that one method of calculation may give very good results on a certain engine, but when applied to another engine will not work at all, proving that the method is wrong. In the case of the four-cylinder engine, for example, the writer does not find any figure introduced for the moment of inertia of the reverse gear, or for the flexible coupling; these masses are certainly not to be neglected, particularly that of the reverse gear.

The last paragraph of the paper is also quite interesting since it points out the difference of the torsional vibrations at the brake and at the flywheel end. This condition is only too natural, considering that a crankshaft is nothing but a spring under torsion, and under this condition there must be a much more even torque at the flywheel end than at the crank farthest away from the flywheel. This condition is the reason why the drive for the camshaft on a properly designed Diesel engine is taken off at the flywheel end, and not at the end away from the flywheel. This fact is very important for a precise fuel injection, which is in turn controlled by the camshaft.

J. Ormondroyd.[1] Mr. Lack and Mr. Jahnke have attacked a very difficult problem in an elementary way. The theory advanced in the early part of the paper is in no wise adequate for the problem in hand. The equation in Pars. 8 and 30 hold for the linear motion of a body free in space with no elastic restraints, acted upon by a force varying in a simple harmonic manner, and having an amplitude T. In this case we do not have such a free body, nor are the forces acting upon it of a simple harmonic nature.

It is difficult to see how the equation

$$F = \frac{60}{2\pi}\sqrt{\frac{I_p G g}{\Sigma(LWr^2)}}$$

given in Par. 11 arises. The same remark applies to the graphical method of finding $\Sigma(LWr^2)$ given in Par. 59. A reference to the complete development should be given. The fundamental assumption involved in the above equation is that a shaft bearing any number of flywheels has only one natural frequency. This is very far from the facts. By analogy to Lord Rayleigh's graphical method for determining the lowest linear natural frequency of a loaded shaft, the writer assumes that the method outlined above

[1] Motor Engineering Department, Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa.

also determines the lowest natural frequency. But the simplest engine tested is a system with three degrees of freedom and therefore has three natural frequencies.

Damping forces do not influence the natural frequencies to any great extent when they are of the order of magnitude encountered in machinery. This refers to a contrary statement made in Par. 15.

The formula for irregularity of angular velocity given in Par. 30 holds only for simple harmonic motion — and this type of motion is not present in the case of internal-combustion engines. From the records printed, it is possible to show a δ of any value up to infinity by the method of slopes by choosing other points than the authors choose.

The method of finding the equivalent length is obtained from a paper written by Dr. Geiger, the inventor of the instrument. This paper has been criticized by Dr. S. Timoshenko in two papers delivered before the Society at the Annual Meetings of 1922 and 1923.[3] No attempt was made to check the results obtained in Par. 29; but Timoshenko's papers show that it is not as simple as indicated in the equations of Par. 29. It cannot be generalized but each case must be calculated separately.

The statement made in Par. 13, and repeated in Par. 50, that flywheels placed in the nodes do not affect the vibrations loses sight of the fact that such an inclusion changes the problem entirely; each additional flywheel adds a new natural frequency and changes the amplitude of all resulting vibrations from those of the original system.

The statement that the flywheel is always the seat of the most violent vibrations cannot be made so generally as it is in Par. 50. The torsiograph itself, in which the flywheel does not deviate from constant velocity, is an example that the vibration on the flywheel depends entirely on the relationship between the flywheel inertia and the shaft flexibility.

As to the technique of using the instrument on jobs where large fluctuations of speed occur, it is always advisable to use as small a pulley as possible on the engine shaft. The authors have used large pulleys and their records have gone beyond the limits of the instrument.

In Par. 33, it is stated that the flywheel in the instrument will rotate uniformly in spite of vibrations on the pulley. This is true only when $\omega = \infty$. It is practically true when $\frac{\omega}{\omega_c} \geq 4$. ($\omega_c$ = critical speed of instrument.)

The stress calculations do not bring out the importance of stress concentration at the juncture of the webs and cranks. The calculations also include the factor WD^2/g. This equals four times the moment inertia of the body considered. The writer fails to

[3] See Trans. A.S.M.E., vol. 44 (1922), p. 653 and vol. 45 (1923), p. 449.

find the factor 4 taken care of in the equations. Perhaps the stresses calculated are 4 times too large.

To measure stresses with the torsiograph, two instruments must be employed simultaneously.

There seems to be some confusion in the units employed. Although the problem is torsional, linear units are used; this only adds work to the problem and does not give a clear picture of what is really happening. Velocities are mentioned as " inches " in Par. 57. Wr^2 and WD^2 are both used in Pars. 59 and 60, one in " inch " units and with the radius of gyration, and the other in " foot " units with the diameter of gyration. And both are called moment of inertia when in reality Wr^2 is gI and WD^2 equals $4gI$. In Par. 64 ω is defined as " per second " which is unitally true, but incomplete.

The calibration test of the instrument was run on a system shown in Fig. 11, which is entirely out of the instrument's range. The instrument is to be attached to systems so much larger than itself that motions of the instrument will not affect the motions of the system tested. The tested system has moments of inertia of the same order of magnitude as that of the instrument and motor. This makes the system one of four degrees of freedom instead of one, and the natural frequencies and amplitudes must be affected.

Another thing is to be noted about this test set-up. The 508 r.p.m. calculated does not refer to 508 complete rotations of the two-body system per minute. It refers to a disturbing force which is variable (sinusoidally) having a frequency of 508 cycles per minute. If the rotor which actuated the system ran at constant speed at 508 r.p.m. no torsional vibrations would result in the system. Fig. 12 does not appear to be a record of resonant vibrations; such a record is usually made up of oscillations of large amplitude — the separate cycles being very distinct.

Fig. 6 shows the connecting-rod effect with four positive peaks in spite of the fact that the mathematical expression ($Rr^2 2\lambda \cos \alpha \sin^2 \alpha$) for this effect contains $\cos \alpha$ which must have negative values in the second and third quadrants.

THE AUTHORS. Three sentences have been added to Par. 29 to meet the point raised in the second paragraph of Mr. Schreck's discussion.

In regard to Par. 59, all graphical determinations of the node were checked analytically; the numerical values were figured with the slide rule and converted into even numbers.

In the two- and four-cylinder engines no serious critical speeds were recorded with the torsiograph.

In the four-cylinder engine the moments of inertia of the reverse gear and of the light flexible coupling have such small values that the masses were not considered separately. The error resulting from this neglect cannot be very considerable, because these masses

lie near the node. Of much higher influence are these parts to the regularity of the engine due to their flexible connection with the shaft.

In regard to criticisms made by Mr. Ormondroyd: The determination given by the first part of the paper up to Par. 30 is a very brief extract from the well-known theory of vibration of simple harmonic nature. It is impossible to give in a brief paper the entire development of this theory with all necessary arguments.

The authors are aware of the fact that a loaded shaft system with a higher degree of freedom has more than one natural frequency. They did not consider the frequencies of higher order.

Par. 15: It should not be understood that damping effects are influencing the numerical value of the natural frequency; but damping effects are influencing the amplitude of vibrations.

Par. 30: The assumption of a simple harmonic motion is not absolutely correct for all fluctuations recorded in Diesel engines. A sine curve represents the effective motion very closely and is approved by other reputable authors. The coefficient δ of irregularity is determined by the *steepest* tangent to the fluctuation curve; therefore, only one value for δ is possible. Some judgment must be exercised in interpreting a diagram.

Pars. 13 and 50: It is correct that flywheels placed in the nodes do not affect the vibrations, as far as this fact is claimed by Pars. 13 and 50; naturally, additional flywheels add new natural frequencies. But no considerable alterations are made necessary in the authors' examples because the critical speeds due to the new additional frequencies range out of the operation speed. Nevertheless, they do not claim to neglect this influence; the effect of additional rotating masses, placed in the node, must be examined in all cases with respect to the new frequency.

In agreement with Mr. Ormondroyd the authors approve that the statement that the flywheel is always the seat of the most violent vibrations, cannot be generalized.

Par. 81: These results prove that the torsiograph drive by the engine flywheel was too big; the figure for δ does not seem correct in this case; therefore, it is preferable to use a small pulley on the shaft to be tested.

Par. 64: In the stress calculation the numerical value of WD^2/g is correct because the moment of the inertia of the flywheel is figured with respect to the diameter D in inches; also, the twisting moment is referred to the diameter with this unit. The figured example was given only to demonstrate the high ratio of additional stress due to torsional vibrations. But the authors accept the statement of Mr. Ormondroyd that for exact measuring of stresses two instruments must be employed simultaneously. The applied method of determination of these stresses is applicable only for a simple system with one rotating mass.

A distinction is made in the paper between the moment of inertia of rotating masses for figuring the irregularity δ due to the flywheel effect WD^2 in lb-ft.^{2}, and Wr^2 = lb-in.^{2} for figuring the frequency of torsional vibrations. Of course, for both calculations the same expression Wr^2 can be applied. It is only necessary to convert the flywheel formula for δ with respect to this unit.

Mr. Ormondroyd's note relating to the test shown in Fig. 11 is right; but this test does not claim any absolute accuracy for numerical values of tests, because it was only a simple demonstration for the existence of a critical speed.

Par. 18: Fig. 6 shows a correct diagram of the connecting-rod effect; it cannot be stated that negative effects occur in torsional vibrations.

No. 1970

FORMULAS FOR COMPUTING THE ECONOMIES OF LABOR-SAVING EQUIPMENT

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The engineer and manufacturer have heretofore been without correct and adequate means for measuring the relative economics of two or more industrial operations or installations of equipment. Proposed new apparatus might indicate a saving in manual labor, but this did not guarantee that it could be profitably substituted for machines or devices in use at the time. Such formulas as existed to determine this latter point were too complicated to be easily applied and generally used.

The present paper develops five simple equations for determining the economics of an installation of labor-saving equipment. They show the maximum investment which will earn simple interest; the yearly cost to maintain the mechanical equipment ready for operation; the yearly profit, in excess of simple interest, from the operation of the equipment; the yearly profit from operation, in per cent on investment; and the years required for complete amortization of investment out of earnings. The method of applying these formulas, the evaluation of their various factors, and the correct assumption of bases, are explained in detail. The calculation of a labor-burden account, as distinct from the ordinary factory-burden account, and the transfer of certain items from the regular burden account to a special productive-labor-payroll account are recommended and explained. Standard forms to systematize investigations are worked out.

THE modern conception of engineering has broadened the engineer's field of activity to include responsibility for economic as well as technical achievements. It is essential, therefore, that dependable and convenient methods for economic research be

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furnished him; also that provision be made in our accounting systems for developing economic information to enable him to predict the results which will follow the adoption of his recommendations.

2 Periodic shortage of industrial labor, its rapidly increasing direct cost, and an indirect cost of considerable magnitude incidental to its employment under modern factory conditions, render accurate methods for analyzing its economic relation to mechanical aids and substitutes a matter of prime importance. Under the relatively simple organization of our industries a generation ago, comparative efficiency estimates, based upon the exercise of personal intuition or judgment, may have closely approximated reality. The relatively complex organizations which present-day conditions make essential usually render such crude processes of estimating inadequate and inaccurate.

3 A survey made for the purpose of finding out what current practice was followed in evaluating labor saved by improved organization or mechanical equipment, brought out the facts that:

1 There is in use no adequate or generally accepted method for estimating the comparative economic value of alternative industrial processes, or for evaluating labor saved

2 Individual practice usually approximates one of two widely divergent methods. The one most frequently employed estimates the value of labor saved at its base or payroll rate, with no contingent addition. The other, more rarely used, but growing in favor because of the palpable inaccuracy of the first, estimates the value of labor saved at the payroll rate plus full factory burden. The range of variation in practice which has been observed between these two methods is as high as 500 per cent.

4 Common procedure having failed to produce any foundation for an adequate system of industrial economic analysis, it becomes necessary, in view of the obvious needs of the engineering profession and of industry, to build up a suitable system. As might be expected, there is not unanimity of opinion as to the requirements for such a system.

5 It is admitted that the basic principles involved have been identified previously, and that there are extant formulas which, when skilfully applied, are capable of solving the economic problems of both engineering and industry. Experience has proved, however, that it is impracticable for any except economic specialists to employ successfully the methods heretofore available. As a matter of fact, a generation, during which their necessity has been appreciated, has not brought them into practical use in industry to any great extent. While it is possible for the engineer to use them, it is improbable that he will in general do so.

Elements for Industrial Economic Research

6 The following elements are considered requisite in making up a system for industrial economic research adapted to the needs of the engineer:

(*a*) Formulas adapted for integrating in a single equation the factors involved in usual industrial economic problems

(*b*) Standardized values for all economic factors which are capable of standardization

(*c*) Standardized methods for developing the values of economic factors which are variable

(*d*) Standard forms to afford a uniform procedure in economic research, and to reduce the clerical work involved

(*e*) Standard forms for summarizing economic data so as to constitute a uniform and thus comparable record.

7 The details of the system should show the entire process of analysis in a manner easily checked, corrected, or modified.

The System Recommended

8 The formulas[3] reported to the Materials Handling Division by a sub-committee during the Spring Meeting at Montreal in 1923 possess advantages which led to their approval by the committee for the following reasons:

(*a*) They are so composed as to develop in five simple equations all facts usually required

(*b*) Unlike other methods available, both debit and credit factors are utilized. Hence the formulas give results in terms of net performance

(*c*) The factors employed generally represent a single element of value. They are composite only where, in current practice, familiar elements are combined and are closely related. Standardized values may therefore be readily and accurately applied

(*d*) They combine flexibility with mathematical exactness in a degree not heretofore possible.

9 In their original form, in deference to established methods of cost accounting, the credit factor "T, yearly saving in fixed charges, operating charges or burden, in dollars," was made to cover indirect costs arising from the employment of both labor and mechanical equipment. It has been found desirable to separate this factor into two elements, as follows:

T_a = yearly saving in labor burden, in dollars

T_b = yearly fixed charges, in dollars, on mechanical equipment employed as a standard of comparison, or which will be displaced.

[3] *Mechanical Engineering*, vol. 45, no. 9, Sept. 1923, p. 564.

10 T_a therefore represents the difference in indirect cost of labor (labor burden) between the two processes which are under comparison.

11 T_b represents the aggregate value of $A+B+C+D$, expressed in dollars, applied to mechanical equipment which is being used as a standard of comparison or which will be displaced.

Formulas for Computing the Economies of Labor-Saving Equipment

12 The formulas for computing the economies of labor-saving equipment are presented for adoption in the following modified form: Let:

Debit Factors:
- A = percentage allowance on investment
- B = percentage allowance to provide for insurance, taxes, etc.
- C = percentage allowance to provide for upkeep
- D = percentage allowance to provide for depreciation and obsolescence
- E = yearly cost of power, supplies, and other items which are consumed, total in dollars

Credit Factors:
- S = yearly saving in direct cost of labor, in dollars
- T_a = yearly saving in labor burden, in dollars
- T_b = yearly fixed charges, in dollars, on mechanical equipment employed as a standard of comparison or which will be displaced
- U = yearly saving or earning through increased production, in dollars

- X = percentage of year during which equipment will be operated
- I = initial cost of mechanical equipment
- K = unamortized value of equipment displaced, less its resale or scrap value

Results:
- Z = maximum investment, in dollars, which will earn simple interest
- Y = yearly cost to maintain mechanical equipment ready for operation (fixed charges)
- V = yearly profit, in excess of simple interest, from operation of mechanical equipment
- P = yearly profit from operation, in per cent on investment
- H = years required for complete amortization of investment out of earnings.

Then

$$Z = \left[\frac{(S+T_a+U-E)X+T_b}{A+B+C+D}\right]-K \quad \ldots\ldots\ldots \quad [1]$$

$$Y = I(A+B+C+D) \quad \ldots\ldots\ldots\ldots\ldots \quad [2]$$

$$V = [(S+T_a+U-E)X+T_b]-[Y+(KA)] \quad . \quad . \quad . \quad [3]$$

$$P = \frac{V}{I} + A \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [4]$$

$$H = \frac{100 \text{ per cent}}{P+D} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [5]$$

Valuation for Debit Factors of the Formulas

13 The debit factors A, B, C, D, and E employed in the formulas are capable of having arbitrary valuations assigned to them when related to definite classes of materials handling or other mechanical equipment. The committee is now engaged in standardizing these values. When this work is completed the use of these standards will avoid the employment of improvised, and consequently erratic, values, and will tend to make analyses, conducted according to the proposed system, of standard composition and hence truly comparable.

Valuation for Credit Factors of the Formulas

14 The credit factor "S, yearly saving in direct cost of labor, in dollars," must of course be developed by a detailed analysis of the processes under investigation. The hourly saving in labor, valued at the base or payroll rate, should be multiplied by 2400 working hours (300 eight-hour days) to develop the saving for a *full year*. This value, with other credit factors, becomes adjusted to the actual conditions when modified by the factor "X, percentage of the year during which the equipment will be operated."

15 The credit factor "T_a, yearly saving in labor burden, in dollars," as already explained, represents the difference in indirect cost of labor required by alternative processes. Details of the indirect expenditures which are elements of this factor are given later in this paper.

16 The credit factor "T_b, yearly fixed charges, in dollars, on mechanical equipment employed as a standard of comparison, or which will be displaced," is the aggregate value $A+B+C+D$ expressed in dollars. The total of these factors appears in the formulas as a credit, opposed to the equivalent aggregate of the same factors applied to the alternative mechanical equipment as a debit, expressed in per cent.

Items Which Are Elements of T_a and T_b

17 In the employment of either labor or mechanical equipment under factory conditions there are two separate classes of expense, direct and indirect. Direct expenditures, as a rule, may be readily determined. Indirect expenses are more difficult to identify.

18 One manufacturer, who has made a careful analysis of the indirect expenses incidental to the employment of factory labor

for the year 1923, determines the average ratio of indirect expense to payroll, for his enitre plant, at 58.6 per cent. Since the conditions in this factory were fairly representative of its class, it is evident that any system which evaluates labor, whether labor used or labor saved, at the base or payroll rate, with no addition for indirect expenses, results in substantial error.

19 In arriving at the cost of labor used it is now customary to add to the payroll rate, as overhead expense or burden, a percentage sufficient to cover all indirect expenses. In recognition of the error arising from a failure to make a suitable addition in the case of labor saved, some have adopted the practice of adding for labor saved an amount equal to that added for labor used. It is true that the cost of labor is substantially the same, whether it is labor used or saved. Hence if the overhead, or burden percentage, were composed exclusively of items which are incidental to the employment of labor, it would be, when added to the payroll rate, a true measure of the value of labor, whether used or saved.

20 Unfortunately for such a purpose, however, cost-accounting systems do not usually develop as burden an amount covering simply the indirect cost of labor, because of the equal necessity for prorating to product, in terms of percentage added to productive labor, the direct and indirect expenses of the factory equipment, as well as a variety of other expenses which can be thereby more accurately, or conveniently, distributed into the cost of product. As a consequence factory burden, as currently accounted, covers not only the indirect cost of labor but also taxes, insurance, depreciation, maintenance, and all other fixed charges against plant and equipment; the cost of power, heating, lighting, etc.; and the payroll pertaining to all labor classed as non-productive. Accordingly we have, by using present accounting methods for our purpose, two primary sources of error in the burden account when setting a value on labor saved:

(*a*) The burden account is excessive as a measure of the indirect cost of labor, because it contains much that is irrelevant to labor costs

(*b*) A portion of the productive labor has been classified as non-productive, and has been charged to the burden account instead of to the productive-payroll account; that is, the productive-labor account does not cover all of the labor which is truly productive.

21 As an expedient in accountancy it is entirely rational to classify as non-productive certain labor which is in fact productive. The work of yard men, truckmen, crane operators, and all others who handle materials cannot in some instances be charged directly to product. Hence such labor is properly classified as non-productive for accounting purposes, and is made a part of factory burden. It is irrational, however, to employ a factory burden thus composed

as a measure of the incidental cost of labor. It is equally irrational merely because certain labor has been given a fictitious classification for accounting purposes, to give it a similar fictitious valuation in an efficiency study.

22 In a plant or department having a burden rate of 100 per cent a workman who receives 70 cents per hour cannot properly be said to cost $1.40 per hour when operating an engine lathe upon product, and but 70 cents per hour when repairing this engine lathe. Charging the labor at the varying rates may produce accurate final results in accounting the cost of product, but it fails completely as a means for developing the true cost of the labor performed, or saved.

23 From the foregoing explanation it becomes clear that to meet the needs of the engineer in arriving at a value for labor saved by a revision in processes, or by improved mechanical equipment, a new factor must be developed by the cost accountant which shall include all items of expense incidental to the employment of labor, and which shall include all items of the factory burden which are irrelevant to the employment of labor. It is suggested that the new factor be differentiated in name, as in composition, by calling it "Labor Burden."

24 As a practical illustration of the relation which labor burden may be expected to bear to factory burden, a comparison is given here, developed by a manufacturer according to the methods above suggested, and based upon the conditions prevailing in his plant for the year 1923.

$$\text{Factory burden} = \frac{\text{Total of factory-burden account}}{\text{Total of regular productive payroll}}$$

$$= \frac{\$417{,}036.70}{\$280{,}585.05} = 148 \text{ per cent}$$

$$\text{Labor burden} = \frac{\text{Total of labor-burden account}}{\text{Total of special productive payroll}}$$

$$= \frac{\$170{,}757.71}{\$290{,}947.51} = 58.6 \text{ per cent}$$

$$\text{Relative ratio} = \frac{\text{Labor burden}}{\text{Factory burden}} = 39.6 \text{ per cent}$$

25 The figures given are an average for the entire plant, and were developed for the purpose of showing the wide difference between factory burden and labor burden. Current practice in developing separate burden values for each department of the factory, while equally applicable in the case of the labor burden, is not considered equally necessary.

26 Factory burden may vary widely. Variations exceeding 400 per cent between departments of a factory or in different industries are not unusual, owing to the wide differences in the relation which fixed charges (such as taxes, insurance, depreciation, maintenance,

TABLE 1 SEPARATION OF FACTORY BURDEN AND LABOR BURDEN

Indirect manufacturing expenses	Factory-burden account: Non productive payroll (Col. No. 1)	Factory-burden account: Other expense (Col. No. 2)	Factory-burden account: Total regular burden (Col. No. 3)	Labor-burden account: Total labor burden (Col. No. 4)	Labor-burden account: Transferred to productive payroll (Col. No. 5)
General					
1 Supervision, cost, employment, production, watchmen, cleaners, overtime, stores office, etc.	$69,552.24	$744.39	$70,296.63	$59,752.13	
2 Supplies, misc. expense, etc.	25.33	10,307.64	10,332.97	8,783.02	
3 Accident compensation, hospital, sick and death benefit		13,665.25	13,665.25	11,615.46	
4 Purchasing, receiving, unassignable, freight, express and cartage	6,890.64	1,878.78	8,769.42		
5 Telegraph, telephone, traveling, etc.		8,040.87	8,040.87		
6 Insurance, taxes, depreciation		9,809.76	9,809.76	3,269.94	
7 Maintenance	1,586.58	1,353.29	2,939.87	2,498.88	
8 Miscellaneous labor	736.11		736.11		$736.11
Total	$78,790.90	$45,799.98	$124,590.88	$85,919.43	$736.11
Departmental Service					
1 Small tools, maintenance, and replacements to tools and jigs, etc.	$13,608.61	$18,100.87	$31,709.48		
2 Auto truckers, helpers, stock room	12,440.89	3,109.95	15,550.85	$3,109.95	$12,440.89
3 Tool crib	7,284.26	72.51	7,356.77	7,356.77	
Total	$33,333.76	$21,283.33	$54,617.09	$10,466.72	$12,440.89

Power Plant					
1 Operating labor	$6,682.14		$6,682.14		
2 Fuel and other supplies		$15,730.11	15,730.11		
3 Maintenance	905.41	983.50	1,888.91		
4 Insurance, taxes, depreciation		3,886.10	3,886.10		
Total	$7,587.55	$20,599.71	$28,187.26		
Direct Departmental Expense of Machine Shop, Assembly Shop, etc.					
1 Foremen, despatchers, cleaners, overtime	$35,702.11	$665.62	$36,367.73	$36,367.73	
2 Inspectors, laborers, stock, distribution, etc.	36,862.26	2,496.00	39,358.26	2,496.00	$36,862.26
3 Defective work	745.45	18,595.19	19,340.64	19,340.64	
4 Defective purchased material and shop supplies		19,053.25	19,053.25		
5 Insurance, taxes, depreciation		39,556.33	39,556.33	13,458.58	
6 Maintenance, land and buildings, machinery, tools, patterns, etc.	17,647.86	6,084.77	23,732.63	2,708.61	
Total	$90,957.68	$86,451.16	$177,408.84	$74,371.56	$36,862.26
Direct Departmental Expense of Drafting Room, Tool Room, and Pattern Shop					
1 Supervision, foremen and other labor	$22,506.75		$22,506.75		
2 Supplies		$4,014.04	4,014.04		
3 Insurance, taxes, depreciation		4,552.54	4,552.54		
4 Maintenance	659.92	499.38	1,159.30		
Total	$23,166.67	$9,065.96	$32,232.63		
Total manufacturing expense	$233,836.56	$183,200.14	$417,036.70	$170,757.71	$50,039.26
	Footing No. 1	Footing No. 2	Footing No. 3	Footing No. 4	Footing No. 5

(Continued on page 534)

TABLE 1 (Continued)

DEVELOPMENT OF FACTORY BURDEN

Total of all factory payrolls	$514,421.61
Less total of non-productive payroll, Footing No. 1	233,836.56
Difference = regular productive-labor payroll = Footing No. 6	$280,585.05

$$\text{Factory burden} = \frac{\text{Total of factory-burden account, Footing No. 3}}{\text{Total of regular productive payroll, Footing No. 6}} = \frac{\$417{,}036.70}{\$280{,}585.05} = 148 \text{ per cent}$$

DEVELOPMENT OF LABOR BURDEN

Regular productive payroll, Footing No. 6	$280,585.05
Less drafting, tool, pattern, and plant-improvement payrolls	39,676.80
	$240,909.25
Plus items transferred to productive-labor payroll, Footing No. 5	50,039.26
Total of special productive-labor payroll, Footing No. 7	$290,947.51

$$\text{Labor burden} = \frac{\text{Total of labor-burden account, Footing No. 4}}{\text{Total of special productive payroll, Footing No. 7}} = \frac{\$170{,}757.71}{\$290{,}947.51} = 58.6 \text{ per cent}$$

Ratio labor burden to factory burden = 39.6 per cent

etc., upon buildings, machinery, and tools) bear to the productive-labor payroll. Fixed charges upon plant represent an important element of factory burden, hence relatively high factory-burden rates follow when the relation of investment in plant and equipment to productive payroll is high. They are relatively low when the relative investment in plant and equipment to productive payroll is low.

27 The items of indirect expense which go to make up labor burden, being independent of the physical surroundings under which the labor is performed, and arising purely

out of the organization and related factors necessary for the direction and control of labor, may be expected to remain fairly uniform under a given type of organization. Such differences as are to be expected would arise principally through a varying relation between the relative elaborateness of organization conventional in a given line of business, and the relative wage scale. Experience seems to indicate, however, that the more highly skilled and hence highly paid trades generally demand a more elaborate and hence more costly organization for their guidance, and that accordingly the *percentage* which labor burden bears to payroll may not be subject to much variation in correctly organized industries of a given class.

28 The development of the exact labor burden pertaining to an industry, or to any department of an industry, is a simple matter in case extreme exactness is desirable. Labor burden may be developed from the same accounting which develops factory burden, values of the items which are chosen to compose the labor burden being carried to an extra column so as to form a separate total.

29 An important difference in the functions performed by factory burden and labor burden should not be overlooked in choosing items which are to be included in the latter. Factory burden represents an average ratio which a certain class or group of expenditures bears to productive labor in making up the cost of a given product or service. Hence a high degree of exactness in reflecting actual factory conditions *at the time the work is done* is essential as a means of developing actual factory costs at that particular period. Labor burden also represents an average ratio which a restricted class or group of expenditures, incidental to the employment of labor, bears to productive labor. Since those items of expense are already elements of factory burden, they have no further significance in the accounting system except as a guide for the future in estimating the prospective cost of similar elements of expense, under modified conditions. Accordingly a high degree of present exactness is not essential, whereas the closest possible approximation to average conditions which may be expected *through a series of years* is desirable.

30 The common expedient of rejecting in the accounting an item of economy because it cannot be realized immediately is not applicable in making up labor burden. The broad principle that, in the long run, incidental expenses will rise and fall proportionately with the productive payroll, becomes applicable.

Items Recommended for a Labor-Burden Account in the Metal Trades

31 When making up a labor-burden account, variations will be necessary to meet the requirements of different kinds of industries. The one here presented is fitted to the needs of the metal trades.

The Materials Handling Committee will compile lists for other trades as rapidly as possible.

Entire cost, with the exception of percentage assessed to non-productive departments	Factory supervision
	Foremen and assistants
	Employment department
	Factory accounting departments
	Factory stenographers and clerks
	Safety department
	Stores office
	Tool storage
	Production department, including all subdivisions
	Traffic department
	Employees taking inventory
	Extra compensation for overtime, etc. (if accounted as an item of burden)
	Watchmen, gatemen, and ushers
	Instruction to apprentices
	Cleaners, oilers, and porters
	Cleaning and miscellaneous work by productive workmen
	All other similar expenditures.

32 Factory supervision should not be construed to cover supervision of the drafting room when it is a factory department. Foremen and assistants in non-productive departments such as tool room, pattern shop, development work and plant-improvement work, should be excluded from the special burden account.

Entire cost	Accident compensation insurance
	Sick benefit and employees' relief
	Employees' death benefit
	Shop vacation allowances
	Maintenance of hospital, cafeteria, club, and welfare activities
	Gratuities
	Defective work and other losses due to errors
	Complaints and correction of product
	Factory office supplies
	Miscellaneous factory supplies
	All other similar expenses.
Part cost	Building rental
	Factory insurance
	Fire department
	Factory taxes
	Maintenance of land and factory buildings
	Factory depreciation.

33 In assessing the percentage of insurance, fire department, taxes, maintenance, and depreciation, that percentage of the total cost should be included in the labor burden which will represent the *proportion of the occupancy of the building by workmen* as compared to other types of occupancy.

34 Since the regular productive payrolls do not ordinarily cover all of the labor which contributes toward production, the following classes of labor, often accounted as non-productive as far as they are included in the regular burden account, should be transferred from the regular burden account to the special productive-labor payroll:

ITEMS RECOMMENDED TO BE TRANSFERRED FROM REGULAR BURDEN ACCOUNT TO SPECIAL PRODUCTIVE-LABOR PAYROLL

Entire payroll	Railway crews
	Materials-handling and miscellaneous labor
	Stores department materials-handling labor
	Auto truckers and helpers
	Inspectors
	Receiving and disbursing raw stock
	Crane and elevator operators
	Shipping department
	All other similar labor.

35 The following classes of labor, as far as they are included in the regular productive payroll, should be deducted in order to develop the special productive labor payroll:

ITEMS RECOMMENDED TO BE DEDUCTED FROM REGULAR PRODUCTIVE PAYROLL TO DEVELOP THE SPECIAL PRODUCTIVE-LABOR PAYROLL

Entire payroll	Drafting room
	Tool room
	Pattern shop
	All development work
	All plant-improvement work.

36 The percentage which the labor-burden total bears to the corrected or special productive-labor-payroll total will represent the labor burden in the form of a percentage to be added to all productive labor. This process will develop a general labor-burden percentage. Where the burden accounting is departmentalized, a similar process conducted for each department will develop the departmental labor burdens.

37 A method which has been found convenient for developing labor burden consists in providing additional columns on the analysis sheet so that items may be transferred, as far as desired, from the regular burden account to these extra columns to form

TABLE 2 COMPARATIVE ECONOMIC ANALYSIS NO. 3 (SEE FOOTNOTE, P. 542)

Showing values for Z, V, P, and H as per formulas recommended by A.S.M.E. Committee.

It is assumed that the use of No. 2 equipment is generally accepted practice, while No. 1 is under examination for comparative economic efficiency. The factor K should be given zero value unless No. 2 is in use and would therefore be displaced by No. 1 with a corresponding capital loss.

No. 1 (designated by subscript numeral 1): 1 Semi-automatic turret lathe. Cost, $6100; operating 2280 hours per year; requires 1 operator; hp. required, 5; cost of power, 3 cents per kw-hr.; reduced occupancy of buildings valued at $115.

Operated for Equal Production in Comparison with

No. 2 (designated by subscript numeral 2): 2 Engine lathes. Cost, $4800; operating 4320 hours per year; requires 2 operators; hp. required, 6; cost of power, 3 cents per kw-hr. This equipment is not in use.
Average values. None.

DATA REQUIRED FOR ANALYSIS

No. 1 One turret lathe		1 year = 2400 hours		No. 2 Two engine lathes	
One		Number of units employed		Two	
$* X_1 = 95\%$		Per cent of year employed	(X)	$* X_2 = 90\%$	
$A_1 = 6\%$		Per cent on investment	(A)	$A_2 = 6\%$	
$B_1 = 3\%$		Per cent for insurance, etc.	(B)	$B_2 = 3\%$	
$C_1 = 5\%$		Per cent for upkeep	(C)	$C_2 = 5\%$	
$D_1 = 10\%$		Per cent for depreciation	(D)	$D_2 = 7.5\%$	
Total = 24%				Total = 21.5%	
E_1X_1 =	$250.00	Cost of power, etc.	(E)	E_2X_2 =	$284.00
2280 hr. at 60 cents	$1368.00	Cost of labor	(S)	4320 hr. at 70 cents	$3024.00
80% of payroll	$1094.40	Cost of labor burden	(T_a)	80% of payroll	$2419.20
$I_1(A_1+B_1+C_1+D_1)$ =	$1464.00	Cost of fixed charges	(T_b)	$I_2(A_2+B_2+C_2+D_2)$ =	$1032.00
Total for 95% of yr.	$4176.40	Cost of operation		Total for 90% of yr.	$6759.20
	115.00	Value of increased production	(U)		none
	$6100.00	Cost of equipment	(I)		$4800.00
		Unamortized value of equipment less resale value	(K)		$1320.00

Valuation of Factors

	$A_1+B_1+C_1+D_1$		24 % =	$A+B+C+D$
	E_1-E_2 =	$250.00 − $284.00 =	− $34.00 =	E
	S_2-S_1 =	$3024.00 − $1368.00 =	$1656.00 =	S
	$T_{a2}-T_{b1}$ =	$2419.20 − $1094.40 =	$1324.80 =	T_a
	T_{b2}	=	$1032.00 =	T_b
	U_1-U_2 =	$115.00 − none =	$115.00 =	U
[2]	$Y_1=I_1(A_1+B_1+C_1+D_1)$	=	$1464.00 =	Y
	I_1	=	$6100.00 =	I
	K_2	=	$1320.00 =	K

Solutions

Z = Maximum investment which will return simple interest on investment

[1] $$Z=\left[\frac{(S+T_a+U-E)X+T_b}{A+B+C+D}\right]-K = \$17340.00$$

V = Yearly profit from operation

[3] $$V=[(S+T_a+U-E)X+T_b]-[Y+(KA)] = \$2697.80$$

[4] P = Yearly profit in per cent on investment $P=\frac{V}{I}+A = 50.2$ per cent

[5] H = Years for complete amortization of investment out of profits $H=\frac{100\%}{P+D} = 1.66$ years

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* X is applied individually in Cases Nos. 1 and 2, hence $100\% = X$

the labor-burden account. The values for both accountings are thus developed in parallel. A specimen form for such work is given in Table 1.

Necessity for a Correct Assumption of Bases

38 Experience in the use of the formulas for computing the economies of labor-saving equipment indicates that a correct assumption of bases upon which an economic analysis is to be conducted is equally vital with the subsequent mathematical treatment. The following suggestions are offered as a guide to methods which have been found preferable.

39 (*a*) The valuation for the factor I should cover the complete cost of mechanical equipment, including special tools, dies, jigs, or other special apparatus; also motive power, in case the motor or other driving agency is a part of or direct connected to the equipment; also cost of installation.

40 (*b*) The proposed equipment (called No. 1 Equipment on the analysis form, Table 2) should be assumed to operate for as great a percentage of the standard year of 2400 hours (any other number of hours may be substituted) as conditions will justify.

41 The function of the factor X in the formulas is to separate time during which the equipment under analysis is in operation productively, and therefore profitably, from idle time during which it is under fixed expense but non-productive. Therefore X_1 should be assumed at the closest approximation to 100 per cent which will reflect actual conditions.

42 For instance, suppose that we are analyzing the comparative efficiency of a machine tool which is to operate upon an item of product not required in sufficient quantity to occupy the equipment full time for a standard year, but the machine is capable of operating upon other items of product also, at approximately equal efficiency. The problem may be solved most conveniently by arbitrarily considering the full range of jobs as one, and extending the operating time to a full year for the main work under analysis.

43 As an alternative which will eliminate all errors made in an approximation like the above, each of a group of operations, which will together occupy the machine for a full year, may be analyzed individually, with the value of X_1 developed for each separately. The analysis for each item of the product should then be conducted according to the formulas, except that the percentage representing the value of $(A+B+C+D)$ should first be multiplied by X_1, this product being employed in the equations. The total of all results will then represent the true efficiency for the year's operation.

44 If it is not found possible to make the group of operations cover a full standard year, then a correct result will be obtained by taking the factor $I_1(A+B+C+D)$, multplying it by the percentage of a standard year during which the equipment is idle, and deducting the product from the above-named aggregate efficiency.

45 The factor X_2 should represent the percentage of a standard year during which the equipment to be displaced (No. 2 Equipment in Table 2) would produce an output equal to the proposed new equipment (No. 1 Equipment in Table 2), not, however, exceeding 100 per cent unless operated during a day-and-night run. For day-and-night running the maximum will be 200 per cent, beyond which two or more No. 2 (displaced) Equipments must be assumed. If not operated during a night run, for percentages above 100 it will be necessary to assume two or more No. 2 (displaced) Equipments.

46 For day-and-night operation it is usual to make the valuation for the factors C and D 150 per cent of that employed for a day run only.

47 (*c*) The credit factor "U, yearly saving or earning through increased production, in dollars," in the sense contemplated by the formulas is the valuation, actual or potential, where an increase in production is or will be required and the usual expedient of extending the plant and employing more workmen is thereby avoided.

48 As an illustration: Two alternative processes show about equal efficiency on the usual comparative basis of equal production. One equipment, however, will be occupied 90 per cent of the time for required production. The other produces the necessary volume of product operating only 60 per cent of the time. Even though not needed at once, the potential extra productive capacity of the second equipment to meet unexpected demand has a certain money value as an insurance of ample product. If the extra productivity can be utilized, and an extra equipment and operator, with incidental building floor space, heating, lighting, etc., are thereby made unnecessary, a very much higher money value for increased production becomes appropriate.

49 In those cases where a strictly limited amount of product is required, and there are no accessory conditions which serve to give a potential or actual value to increased production, it may be necessary to give U a zero value.

Standard Form to Afford Uniform Procedure in Conducting Economic Researches

50 As a convenience to the engineer in conducting economic researches, to afford a readily readable record of the assumptions and processes employed, to standardize the work and results so that they will be comparable with similar researches, to guide

the research along correct lines, avoiding oversight or error, and to save time, the form for comparative economic analyses shown in Table 2[1] is submitted.

STANDARD FORM FOR SUMMARIZING ECONOMIC DATA

51 The summary of an economic research shown in Table 3 reviews the factors which have an economic value, such as:

(*a*) Estimated capital required
(*b*) Estimated yearly profit over alternative
(*c*) Estimated profit in per cent on investment
(*d*) Estimated time required to amortize investment
(*e*) Estimated saving through reduction in labor turnover.

TABLE 3 SUMMARY OF ANALYSIS NO. 3

Estimated actual investment required for No. 1	$6100.00
Estimated excess investment in No. 1 over No. 2, $I_1 - I_2$, or I_1	1300.00
Estimated yearly profit from operation No. 1 over No. 2, excess above simple interest	$2697.80
Estimated value of reduced labor turnover	
Total	$2697.80
Estimated yearly profit from operation of No. 1 over No. 2 in per cent on investment, total	50.2 %
Estimated yearly profit from operation of No. 1 over No. 2 in per cent on excess investment, $\frac{I_1}{I_1 - I_2} P$, total.	235 %
Estimated time required to amortize entire cost of No. 1 out of earnings in excess of No. 2	1.66 years
Estimated time required to amortize excess cost of No. 1 over No. 2 out of earnings in excess of No. 2, $\frac{H}{\frac{I_1}{I_1 - I_2}}$.	0.33 year

This should also include practical considerations which may have no immediate money value, but which may nevertheless provide the ultimate weight of evidence for or against a given choice, such as:

(*f*) Estimated relative flexibility to adapt to unforeseen conditions or types of product
(*g*) Estimated relative continuity of service through relative durability, inherent reliability, or because of labor conditions.

52 The formulas and analyses above presented furnish the engineer with a ready and accurate device by which he can measure the economies of present and proposed operating methods and equipment, and decide correctly and with certainty between alternative proposals.

[1] The analysis number (3) applies to the authors' investigations only and, as a coincidence merely, duplicates an analysis number employed by another author.

[EDITOR'S NOTE: Following this paper by Messrs. Hagemann and Shepard, three others were presented giving applications of the formulas to (1) the purchase and operation of an electric industrial truck to replace hand lift trucks; (2) changes in machine-tool equipment; and (3) labor-saving equipment in road construction. These three papers were published in *Mechanical Engineering*, vol. 47, no. 5, Sec. 2, May 1925, pp. 408-415.]

DISCUSSION

F. O. HOAGLAND.[1] The first attempt of the Materials Handling Division about a year ago to find a solution of the problem of determining the economies of labor-saving equipment attracted the attention of the writer. He recognized that this same useful information could also be applied to machine tools as well as to equipment for transportation, and therefore immediately prepared a form for his own use and for the use of the equipment engineers of the company with which he is connected. Scores of estimates have since then been submitted, using these forms as a basis for the calculations.

These forms also have been used for comparing the costs of certain components of our products, as now being manufactured, with proposals from outside manufacturers. They have definitely proved that in several cases it was economy to discontinue the pouring and machining of gray-iron castings and to substitute parts made of pressed steel by specialists in that field.

The statement in regard to the special production payroll is interesting as it tends toward allocating the various charges where they properly belong.

F. T. SMITH.[2] Of what particular advantage are these formulas to the engineer, or to the public? Why were they derived? Formulas pertaining to the application of new machines for the saving of labor in material handling and economic processes are most important in the development of the industrial field. It is difficult to understand why, with the many other important formulas in common use, something of this nature, which could be used to such great advantage, has not previously been made up or why methods for working them out were not undertaken.

Formulas of many kinds have been derived and are in common use by the scientist, engineer, contractor, builder and estimator, in experimental and research work, for many purposes. Formulas are in constant use by scientists and engineers in problems that range from determining the distance from new planets to the earth to determining the number of turns of a coil of a radio set, and from

[1] Saco-Lowell Shops, Biddeford, Me. Mem. A.S.M.E.

[2] Industrial Department, General Electric Company, Schenectady, N. Y. Mem. A.S.M.E.

designing a tiny package-carrier system to building a huge car dumper. Is it not a logical conclusion that the industrial engineer would be benefited in the application and use of a formula that would show economies as applied to labor-saving devices?

Such an engineer might have problems involving equipment for road construction, the many and various types of machine tools, automatic machinery for quantity production, conveyors for steady-flow production, cranes, hoists, winches, and trucks for material handling in industries, and in both inland and seaport terminals, and the many and varied types of equipment for bulk material such as coal, ore, sand, and limestone. A much wider diversification could not easily be imagined. From this it can readily be seen that the field of the industrial engineer is a broad and highly important one. This, if for no other reason, is sufficient to justify the thought, time, and energy spent to develop something that is useful and helpful.

These formulas can be used to determine values for the many types of equipment and industrial processes. They are generally applicable where economies resulting from the installation of new equipment or the substitution of improved apparatus for obsolete types are to be determined.

In the past few years many new and wonderful devices and machines have been put on the market. These may have been sold by engineer salesmen, who, through good salesmanship, have convinced the buyer. The sale and ultimate installation may have proved entirely satisfactory, but had this same sales engineer had in his file ready for use convincing and condensed formulas to show results, they would have saved him time and energy in presenting the application of his particular type of equipment and would have told the buyer what he was partieularly anxious to know in connection with it.

With something tangible, that will give a definite answer, the engineer approaches the problem before him with confidence. He is able to give a logical, sure, and accurate analysis in a comparatively short time.

At least four things confront the man who is contemplating buying new equipment or making a change, namely, cost, amortization, percentage return on the investment, and increased production. Naturally the buyer hesitates to make any change until he feels certain that he will be benefited thereby. Previously he had no way of knowing what the results would be without using a somewhat laborious method. Now, however, in connection with these formulas, it is possible to reach a logical conclusion.

Labor has always been a variable and unknown quantity. The formulas help to solve its value. In thus determining the economies of processes by new and improved machines or methods showing thereby their value, they also release labor for more important duties. The cost of labor is a worry to the manufacturer, and

probably the future is bringing us a condition where labor will demand easier working conditions and possibly higher wages unless the price of commodities decreases.

The great value of these formulas can only be estimated; their true value will be realized after their usefulness has been demonstrated.

JAMES A. BROWN.[1] There may be substantial agreement that the main point is to use most effectively all the capital invested in a business. But how? The production man is apt to generalize his position to the effect "that any expense is justified if it produces a worth-while economy." Then again, the financial man may hesitate about tying up capital in fixed assets, and he is well within his rights in questioning such expenditures. Obviously a method or formula which harmonizes these divergent viewpoints is much to be desired. Such a formula must be applicable to a business as a whole, as well as to each component building, structure, piece of equipment, or process. To put such a formula in proper perspective it must be remembered that selection involves suitability and cost. The question of suitability is about 95 per cent of the problem, and this is one of technical engineering. Suitability being determined, the very minor part of the problem is one of comparative costs. No cost formulas will in themselves determine suitability although they may influence selection very materially.

In dealing with comparative costs of mechanical equipment, the underlying idea is that most labor-saving devices of a mechanical nature require an investment in machinery or equipment by means of which labor costs may be reduced. But it is the increase in fixed costs that must be balanced against the saving in the direct cost of operation. This general idea is sometimes termed the "economic principle of selection" or "the law of mechanical improvement." It has been stated somewhat as follows: "The quantity and quality of mechanical equipment which it pays to install depends on the amount of use that will be made of it; so also an equipment must earn its costs and profits when in operation, for it certainly cannot do so when idle." This second statement contains the germ of modern costing methods in the emphasis placed on the hours of use as an essential factor in cost comparisons. In its practical application, this idea is of such importance in any analysis of expense distribution that the principle should be thoroughly understood and not confused with form or nomenclature.

Costs must be known before they can be compared and a basis of comparison must be agreed upon. By more or less common consent, the cost of operation per year or total annual cost has been agreed upon as a practical basis. This total cost is made up of two groups of items, namely (*a*) those which remain practically

[1] Mechanical Engineer, New York, N. Y. Mem. A.S.M.E.

fixed and unaffected by the time of operation or by the volume of production; and (*b*) those that vary in some direct way with time or volume of production. For some purposes it may not be necessary to segregate costs in this way, but the relative effect of fixed charges on costs and profits under varying volumes of production can be appreciated only by the construction and study of tentative budgets or production schedules for such different volumes.

The writer believes that the formulas offered by the Committee fall far short of solving in a simple direct way the questions involved in even the simpler cost comparisons.

In the 1925 version of the Committee's formula, while the " total cost of operation " reappears, it is not expressed as a formula and no direct use is made of it, although it is evident that the profit from operation may be obtained directly by subtracting the less amount from the greater. Formula [3] is not only unnecessary but is a cumbersome statement of a simple relation. Another clumsy device is the grouping of the items of cost as " debit " and " credit " items. This is absolutely meaningless, as no functional use can be made of such grouping. A casual examination makes it evident that the ideas expressed as formulas need considerable clarification before they are suitable for general use.

JAMES L. HAYNES.[1] That the Society should eventually take action toward adopting standards for economic performance appears necessary as we review the present facilities for checking or gaging certain steps in industrial processes. For instance, all business concerns of any consequence keep books to show to a cent all income and expenditures. We have had for years and are always expanding the range of standard tests for specified materials, for performance of machinery, etc.; but the matter of predicting or checking the economic advantage or loss from some new plan or device is handled with greater or less accuracy in different organizations and in many cases nothing but intuition is the guide.

By training ourselves to apply a real test to any new proposition, we shall secure a solid basis for discarding ideas that appear to have every qualitative argument in their favor, but which have not any advantage in a sufficiently great degree as to be self-sustaining. The corollary to this is that a quantitative analysis will be of great assistancce in salesmanship when the prospective buyer has become accustomed to recognizing a standard method for checking economic performance.

The field of materials-handling equipment is a most important one in which to begin applying rational tests of economic value. The writer believes, however, that a still broader interest of the Society will be attracted if the application of the formulas is urged in other fields.

[1] Hyatt Roller Bearing Company, Harrison, N. J. Mem. A.S.M.E.

R. C. NEWHOUSE.[1] The formulas presented greatly simplify the methods which have hereto been used for computing the economies of labor-saving equipment, but it would seem that the factor K, as used in the formula, does not show the true economic worth of the new equipment being considered. The unamortized value of the equipment displaced is not truly a factor in determining the worth of the new equipment. From the standpoint of the accounting department, there may be an objection to rendering worthless by replacement a piece of equipment which is carried on the books at a considerable value. However, from a true engineering and economic standpoint, the value at which the old equipment is carried does not enter into the consideration of the saving to be made. The economic test of the value of the new equipment is whether the saving to be effected justifies the new capital investment, and from the cost of the new equipment can properly be deducted the scrap value of the material displaced when making a determination as to the maximum investment in dollars which will earn simple interest.

S. H. LIBBY.[2] The authors have referred to the expense in accounting. We have been accustomed to measure that part of the overhead in the ratio of overhead expense to the cost of direct labor. If that ratio could be dispensed with, much trouble would be eliminated. The fallacy of this ratio has gone so far that arguments are advanced for the increase of the direct-labor payroll. Now we have come to look at things in a different light, and a study of the formulas in the paper will not only give us a means for determining the value of a new machine or process, but willl also enable us to bring old methods up to date. The accountant can use these formulas to determine whether or not the product that is being made is something that should be continued or dropped.

MERRITT LUM.[3] The writer desires to inquire as to why investment is figured on a six per cent basis. For instance, new equipment is wiped out by the depreciation charges, and at the end of the period of amortization the book value of the equipment is zero and the interest on investment zero. If the borrowing rate on money is six per cent, the average cost of the investment will be approximately three per cent.

In regard to the replacement of old equipment with new, there must have been some maintenance charge on the old equipment.

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[2] Consulting Hoist Engineer, General Electric Company, Bloomfield, N. J. Mem. A.S.M.E.

[3] Vice-President, A. W. Shaw Company, Chicago, Ill. Assoc. A.S.M.E.

Inasmuch as a maintenance charge is set up for the new equipment, it would seem logical to set up a credit covering what would have been maintenance cost on the old equipment.

A. E. HOLCOMB.[1] The writer understands that, in comparing two types of equipment, where the substitution of one kind for the other does away with the part of the labor formerly required, the overhead charge against the preceding type is to be deducted in addition to the cost. It would seem, however, that there are certain items of overhead that could not be fairly deducted. For instance, if the company maintains a hospital, or if superintendence or similar charges are included in the overhead, the cost of such items will continue regardless of the change in the type of equipment and the elimination of part of the labor. Consequently the overhead applying to such items could not be eliminated and, it would seem, must be retained in making a comparison.

JAMES A. SHEPARD.[2] It is the object of this paper to direct attention to the lack in our industrial accounting systems of provision for developing economic information required in the work of the engineer, as prescribed by our present conceptions of engineering; also to outline in principle, and to some extent in detail, methods which are considered suitable to meet present requirements.

Throughout industry, a realization of the relative magnitude to which incidental or indirect items of cost will total has generally been belated, and has probably not yet become a matter of common knowledge. Overhead or burden, while generally recognized as an item of cost, apparently has been accepted by some in a half-hearted manner; less through a genuine appreciation of it as a legitimate and unavoidable element in cost than through inability to argue down the theorists who defend it as just as real and necessary as the most obvious direct expenditures.

Fortunately for the stability of industry, those who still entertain that one-time prevalent heresy have become an inconsiderable minority. The extent of incidental or indirect cost of employing labor under factory conditions is, unfortunately, not yet generally recognized.

The value of good working conditions in our factories is admitted by employers of labor, and very generally throughout industry, employers are supplying more adequate buildings than formerly, better facilities for admitting daylight, better means for artificial illumination, better heating and ventilation, shower baths, rest rooms, lunch, and cafeteria facilities

[1] Sales Engineer, Koehring Company, Milwaukee, Wis.
[2] Authors' closure.

Our production systems, and the necessity for employing operatives who are unskilled except for ability to perform some special process, have brought into the factory an increase in the supervisory force, accessory workmen, and clerical employees in comparatively large numbers. The factory hospital, nurse service, medical supervision, safety department, accident compensation, life insurance, better housing, and community welfare have all become recognized attributes of the factory, either without full realization of the percentage which their total bears to the payroll, or at least, if known, without bringing such expenditures into certain of our economic calculations.

Mr. Hoagland's discussion is of particular interest on account of his reference to the special productive payroll and, by inference, the labor burden. Without these two factors no accurate accounting of the value of labor saved seems possible. Unless we are able to arrive at a true valuation for labor saved, no adequate valuation can be placed upon mechanical substitutes for labor or other means for its conservation.

Mr. Smith makes reference to one of the important functions of the proposed system for economic analysis. It is believed that the methods suggested will prove practicable for general use by engineers in testing their various conceptions, and as an accompaniment for their recommendations, serve as unassailable evidence of the value of such recommendations.

In the last analysis the problems of industry are economic; industrial technology becomes of value in whatever measure it contributes toward safety, service, and economic advantage. The work of the engineer must contribute in full measure to such ends, otherwise he may not, as an individual, except to participate amply in its material rewards, or the honor attendant upon worthy accomplishment.

Mr. Brown's discussion does not seem to be entirely germane to its subject. In practical engineering, a determination of the process which will develop maximum theoretical efficiency may be of little or no use, where the alternatives, as is often the case, involve a wide difference in the initial investment required. The temptation to minimize capital outlay for equipment is usually irresistible except for a comparatively wide difference in efficiency in favor of the equipment demanding the greater capital outlay.

One can easily agree, however, with his contention that the subject is a large one, and add, perhaps, that it is too large to permit engineers generally to acquire an exhaustive knowledge of all of its details; it is hoped, therefore, that as appears to be the case, the general temper of engineers will be to welcome a codification of those practical features which will be required for ordinary uses. If we may judge from the results of the survey which has been made to develop current practice, a system for economic research having faults far in excess of those which are

alleged, might still, if generally employed, be of great value as compared with any alternative available.

Since Mr. Brown is obviously in a critical mood, all who have been connected with this matter are to be congratulated that nothing of importance adverse to the proposed system has been brought out, except a personal difference of opinion as to procedure.

It has been the belief of members of the committee having the matter in charge, that, as Mr. Haynes suggests, the formulas for computing the economies of labor-saving equipment are widely applicable in industry, outside the bounds of materials handling.

Mr. Newhouse raises a natural and pertinent question as to the desirability of employing a valuation for K, unamortized value of old equipment, even when equipment which is to be replaced is still carried upon the books at considerable value. It is perhaps a debatable point concerning which authorities will disagree.

Were one confined to a single alternative, it would probably be preferable to neglect any unamortized value in the old equipment, although some would then perhaps consider that the formulas gave an incorrect result. The use of K in the formulas at the book value of displaced equipment at least tends toward conservatism in the results shown. For those who approve the alternative, a zero valuation becomes available.

Practically the same considerations have influenced the committee in the matter of the factor A, percentage allowance on investment. Probably the weight of sentiment is against charging interest on invested capital as a part of production costs, nevertheless we are far from unanimity of sentiment on that subject. The formulas afford an option which may be exercised in favor of charging interest as an item in cost, or through a zero valuation, against such a practice.

Owing to the problems which are encountered when interest and depreciation are charged against mechanical equipment coincidentally (by which the gradual amortization of the investment through depreciation really represents an annually diminishing cost for interest), a slight error in the direction of conservatism will result through the use of the formulas. Many companies prescribe, however, that new equipment must earn a certain percentage of interest on the capital required, even when the general accounting system does not so treat invested capital. Absolute accuracy is not always worth while, otherwise we should sometimes be dealing with an innumerable number of decimal places.

The foregoing applies to the first question raised by Mr. Lum in his discussion concerning the proper charge for interest. As a proposition in accountancy, he is right in assuming that interest will run against the full purchase price of equipment during the first year only.

The Formulas for Computing the Economies of Labor-Saving Equipment have been composed, and are justified, as a means for making economic predictions in advance of actual practice. In this field there appears to be no practical alternative. Accountancy, on the other hand, is an art for developing economic results subsequent to actual practice. It has been found that material error results by an attempt to use either system in the field to which the other has been adapted. That fact has been referred to in the paper.

Assuming, therefore, that the major use for the formulas will be as an economic test for a prospective installation of improved equipment, that test must presumably measure with some rule, such as, first, a given minimum rate of interest on the new capital required; and second, substantial profit in excess of that minimum interest. After installation accountancy will develop refinements in efficiency which will not be important in a preliminary study which must necessarily deal in approximate but safe estimates.

The specimen analysis No. 3 in Table 2 of the paper allows exactly the same percentage for maintenance charges for both old and new equipment, thus answering the second point which Mr. Lum raises.

The answer to Mr. Holcomb's discussion will be found in the paper, at least by inference. It is true that saving labor will not immediately reduce a certain class of overhead charges. The expenses for hospital, or superintendence, referred to, may not reduce at once, neither will they increase at once, even with an increase in labor; hence, in the language used in the paper, "The common expedient of rejecting in the accounting an item of economy because it cannot be realized immediately, is not applicable. . . . . The broad principle that, in the long run, incidental expenses will rise and fall proportionately with the productive payroll becomes applicable."

Since economic analyses, as made by the engineer, must represent average conditions during the lifetime of an installation, the language of the paper is again confirmed: "A high degree of present exactness is not essential, whereas the closest possible *approximation* to average conditions which may be expected *through a series of years* is desirable." It becomes an obligation upon management to see to it that the charges for hospital, superintendence, and all other items of incidental expense are kept proportional to the size of the working force. Then the assumptions employed in the formulas in predicting the probable costs in advance of practical operations will come true.

No. 1971

THE ENGINEER AND CIVILIZATION

By W. F. Durand, Stanford University, Cal.
President of the Society

IF WE MAY ACCEPT the estimates of archæologists, it is now some hundred thousand years or more since man began to manifest, in some marked degree, those characteristics which have served as the foundation upon which the superstructure called civilization has been erected.

Throughout this long evolution, three great principles or lines of action have stood out as the determining factors. These are specialization, coöperation, and the utilization of the resources of nature. These are, in fact, all interrelated. No one is independent of the other two.

So long as the unit of life is the individual, or at most the family, there can be but small progress toward a state of what we term "civilization." So long as a man or a family must do all the things necessary for his or its life, no one thing can be done with super skill or super excellence, and progress must needs be slow. The acquirement of super skill or of super excellence means specialization, concentration, and the limitation of activity to a relatively narrow line of endeavor. Only by this pathway can progress be made. But if the individual is to limit his own activity to a narrow line of endeavor and give to this field of work his entire time and effort, he will obviously be in no position to provide for himself, at first hand, more than some few of the things needful for life, and he must perforce depend on others for those things which he cannot supply himself. This means that he must contribute from the results of his own skill to the needs of his neighbors, and that he must count on receiving from them, in return, the products of their special skill to supplement his own lack. Thus is born the principle of coöperative effort, and out of this flows, of necessity, the commercial institution of barter and exchange.

Presidential Address at the Annual Meeting, New York, November 30 to December 4, 1925, of The American Society of Mechanical Engineers.

The Beginnings of Material Civilization

If we seek to place a finger upon the very beginnings of what we may term "material civilization," we shall find it at that point where some race or community of men of primordial stamp and character began to realize and enjoy the advantages accruing through the utilization of some combination product of human skill directed upon a raw material of the earth. For, with the growth of such a condition, there would come a premium on human skill, and thence an inducement for specialization and concentration on some particular line of human endeavor. And with this must come the whole train of specialized human effort, coöperation, and mutual dependence as foundation blocks for the building up of a civilization.

Again, if we seek to place a finger upon the time, place, and circumstance when, where, and how such a condition first came into evidence as a guiding factor in human evolution, we shall apparently find at least one such point in central and southern Europe during the old stone age, and in connection with the utilization of flint arrow and spear heads for the chase, and of other crude stone implements for the simple and homely arts of that far-away time. Man had placed his foot on one of the first rungs of the ladder of progress toward civilization when he found a way of adapting the crude stone fragments lying about him to the needs of life. Definite and material advantages accrued as a result of the use of such implements fashioned from these resources of nature — certainly among the first of such resources to be utilized for other than direct food purposes.

To take, for a moment, a more narrow view in point of time, consider the culmination of this age in the so-called Solutrian phase of culture, and located, in time, perhaps twenty-five thousand years ago.

In that age there might have been seen, about the wonderful limestone caves of southern France, communities of the people of the age, living in or about the caves and practicing the arts of the time. The food supply came primarily from the chase and the flint arrow head and spear head played here a most important part. In this age the art of fashioning flint arrow and spear heads seems to have reached its culmination. In no age before or since have there been produced their equals in beauty and perfection of form, proportion, and finish.

Now there seems to be good evidence that the peculiar skill which was required to shape these beautiful examples of chipped-stone art was by no means the common property of all the men of a community. Rather was it, presumably, the property of a limited few, or perhaps of one, who would thus, by force of circumstance, become the arrow- and spear-head maker for the community. Here then perhaps may we discern the very *first*

example, in the development of our civilization, of a clearly defined specialization or differentiation of function in a community of human beings. And with such specialization there must come coöperative relations, barter, and exchange, and thus the foundation for a modern state of society.

Is it too forced a use of language to call this ancient arrow-head maker an industrialist, perhaps an inventor, and an engineer?

It is of the essence of the work of the industrialist that he shall organize, plan, and carry forward undertakings calculated to provide, often in quantity and through repetition processes, the material things of which civilization has need.

It is of the essence of the work of the inventor that he is able to recognize some need in the arts of civilization and to find a way to supply such lack.

It is of the essence of the work of the engineer that he deals in a large way with the constructive materials of the earth and with the energies of nature — all to the end of useful service in the arts of civilization.

The Arrow-Head Maker the Prototype of the Industrialist and Engineer

Surely we can discern in the work of this ancient arrow-head maker and in his relation to the community life of that day the clear beginnings of those functions which, in more recent times, we designate as those of the industrialist, the inventor, and the engineer. Indeed, the distinction between these terms is often arbitrary, and the same individual may be, and often is, any one, two, or all three of these at one and the same time. If, then, we may designate, in a large way, this field as that of the engineer, we may fairly say that far back in those prehistoric times, easily 20,000 to 30,000 years before we have even the beginnings of recorded history, we may clearly discern the germ and the essence of the exercise of that function in society which we, as engineers of a later day, consider our own peculiar responsibility.

But the arrow-head maker of the old stone age by no means stands alone as a prototype of the modern engineer. There is his fellow, perhaps of a somewhat later time, who invented the art of weaving, especially as expressed in crude basketry, and thus gave to man, perhaps, his first container or carrier. And again, not long subsequent, that ceramic engineer who discerned, perhaps first through the burning of a clay-lined wickerwork basket, the influence of fire upon clay, and thus laid the foundation of the great modern ceramic industries which have come to mean so much in our own day, and the development of which can be so clearly traced down through the ages. And again, what of that engineer-scientist-inventor of 20,000 years ago, perhaps, who gave to the world of that day the bow and arrow with its wonderfully

effective utilization of the laws of mechanics regarding the transformation of energy? We may, in fact, see in play here the entire sequence of invention and engineering science.

Primitive Utilization of Mechanical Laws

All invention must start with the recognition of a lack or need, and following this there must be some mental vision of a manner in which this lack or need may be supplied. The invention must be subjective before it can become objective. There must be in the mind of the inventor some mental picture of the means whereby the lack may be filled before it can become clothed in material form. And so some one, in some way, in those remote times caught some realization of a need or insufficiency in the weapons used in the chase, and following upon such recognition he must have gained some mental picture of a way in which the need could be met and the insufficiency supplied. And so was produced the combination of bow and arrow, with its wonderfully effective means for receiving the organic energy of the bowman as he pulls the arrow and flexes the bow, and for storing it in potential form, expressed as the potential energy of a distorted elastic system. And then, at the instant of his choice, with the arrow released by the fingers, this stored energy becomes transformed into kinetic energy expressed in the flight of the arrow toward its mark. Truly a wonderful series of energy transformations, and realized with marvelous efficiency and perfect adaptation to purpose!

The Great Antiquity of the Engineering Profession

Do we, as engineers of the present day, stop to remember that whenever, in our modern mechanism, we utilize a spring, no matter of what form or character, we are simply following along the path marked out by this great inventor-engineer of the long ago? Do we not, as did he, simply use the distortion of an elastic system and its release as a means for storing and transforming energy? And again, in this same category of scientist-engineer, there is the inventor of the sling. This ancient military engineer or huntsman found a way of utilizing the kinetic energy of a rapidly rotating material system. We do the same with the flywheel, and, so far as basic principle is concerned, we are simply again following along the pathway first worked out by this engineer-inventor of a vanished race and of a departed age.

And again, have there not been naval architects and shipbuilders since the days of the log dugout — a period of not less than 15,000 years? Have there not been bridge builders since our ancient forebears — no one can guess how many ages ago — first learned to throw a log across a stream, or to stretch, suspension-wise, cables made of twisted vines or other like material?

And have there not been metallurgists and metallurgical engineers since the days of Tubal Cain, who, as we are told in Genesis, was "an instructor of every artificer in brass and iron?" And have there not been great structural engineers and builders who for 6000 years past at least have been enriching the earth with the products of their genius and skill — as witness the wonders of Nineveh, Babylon, Egypt, Greece, and Rome, or again the records of the Maya and earlier civilizations on our own continent?

And so, as we follow the sequence of the utilization of the constructive materials of Nature and of her energies down through the ages — from the products of early paleolithic or rough chipped-stone culture to later paleolithic, and then to neolithic or polished stone, horn and bone, and then on to bronze and so finally to iron and steel and thence into the more complex conditions of modern times — we see continuously and clearly the accumulating results of specialization, of coöperative human labor, and of the inheritance, by each successive generation, of the accumulated culture of those which have preceded it.

If I have taken this length of time to examine the early beginnings of the art and practice of engineering, it has been to draw attention to the great antiquity of our profession and to justify, in some degree, the claim, which we may make, of belonging to the first group or guild of society which became distinguished in relation to the community at large by reason of some peculiar or special function or service rendered.

We sometimes consider that engineering and the engineer, in the distinctively modern sense of the terms, do not antedate the invention of the steam engine and the beginning of the utilization of the inorganic energies of nature — those developments which have so revolutionized the material content of our civilization during the past 200 years. The distinction, however, is superficial rather than fundamental; the informing spirit and purpose are the same. The difference results primarily from the enormously increased magnitude and diversity of the material content of our present-day civilization, arising from the utilization of the inorganic energies of nature in lieu of the organic energies of man or domestic animals. It has resulted, naturally, that the engineer of today must be a man of far different training and type of activity than his brother of 300 or 500 or 1000 or 10,000 years ago. But the point which I make is simply that in spirit, in purpose, and in relation to the civilization of the age, the art and practice of the engineer of the present day have their roots far back before the dawn of written history — back in those first movements in human society which gave evidence of the working of some leaven of evolution, and which in 100,000 years or more have led us from the condition of our paleolithic ancestors up and into the condition and circumstance which we now enjoy.

Modern Material Civilization a Product of the Engineering Guild

And so, in these latter days, we have come into a state of civilization, so called, of which the objective content, with its richness and complexity, is the product of this guild of those who work with the materials of the earth and with the energies of nature. But so commonplace has all this become that the world accepts us and we accept ourselves as part of the scheme of things with little thought of the extent to which this work of the engineer, in the broadest sense of the term, permeates our entire community life.

Suppose we take so humble an object as the slice of toast which appears on our breakfast table. Do we stop to consider the extent to which engineering and the engineer have entered into the vast coöperative enterprise necessary to the production of so commonplace a result? Thus we find that the soil on which was grown the wheat was turned up by a gang plow drawn by a tractor. The grain was sown, harvested, winnowed, and bagged by machinery driven by power. The bags of grain were transported to a rail shipping point by motor truck and thence by rail to a wheat center. The entire program of milling, transport of flour to bakery, mixing, kneading, baking, and delivery at the kitchen door is again realized through the use of power and energy. And finally, its preparation on the electric toaster is an expression of the utilization of power in its very latest and most versatile form.

And again, what of the design and construction of all these mechanisms and agencies — the plow, the reaper, the motor truck, the locomotive and the freight car; the rails on which they run, the bridges, trestles, and tunnels over and through which they pass; the flouring mill, the bakery equipment, the auto-delivery wagon, and finally, the long line of electric machinery, apparatus, and equipment which takes mechanical energy at a far-distant point, transmutes it into the electrical form, and brings it over mountain and valley and across plain to our door, and so to the electric toaster on the breakfast table; there again to be transformed into heat and applied to the bread to render it the more toothsome? And as we thus eat our toast while reading the morning paper, what thought do we give to the manifold energy transformations, the hundreds of mechanisms, devices, and appliances and the thousands of individuals whose joint labor and service have made this simple act on our part possible?

Or again, consider the newspaper itself. The happenings of the entire world are brought by telegraph, telephone, radio, or mail service, assembled within one room, there by typewriter transformed into copy, thence, by the aid of wonderful type-forming machines and technical processes, expressed in the form of records on cylindrical metal shells, and thence, through one of the marvels

of modern inventions, the printing press, transformed into the printed page, moved by auto truck, railway train, and otherwise, and finally laid on our doorstep ready for consumption with the toast. All of these complex operations and processes, as well as the mechanisms and agencies through which they are realized, are again only an outward expression of a combination of the materials of the earth, the energies of nature, and the directive agency of man, and therefore fall within the direct and peculiar field of the engineer.

And as with these familiar examples, so with the entire *material* content of our civilization. The food we eat, the houses in which we dwell, the clothes we wear, the facilities we enjoy by way of travel — on land, over water, or through the air, of trade and commerce with the farthest ends of the earth, of communication with our fellow by written word, telegraph, telephone, or radio, the *material* facilities and means through which we maintain an ordered state of society through government and judicial procedure, the *material* means whereby we transmit the culture of our own age to our children through education, and indeed the *material* background of the organization of our society which makes it possible for select spirits to find time for reflection, for study, for things of the mind and of the spirit rather than things of the body — turn which way we may in the tangled maze of our modern life, and we find on every hand dependence, in some degree at least, on that combination of constructive materials, energies of nature, and the directive agency of man which constitutes the especial field of activity and service of the engineer.

This enunciation of the place of the engineer in the progress of civilization is uttered in no boastful spirit and with no purpose of exalting his services over those of his brothers in arms. The carrying on of our present-day civilization calls for the exercise of a vast number of diverse though interdependent functions. There is no need of an attempt at enumeration. All are necessary, and no one is independent of the others. Rather all are dependent on each and each is dependent on all. If the tiller of the soil, the herdsman, the merchant, the banker, the doctor, the lawyer, are all dependent on the engineer, so is he in turn dependent on them. Life is like some intricately woven web of chain mail wherein each link is necessary to, and in turn dependent upon, all the others. Truly, the ideal of our civilization is that one shall labor for all and all for one.

And so, in the endeavor to mark out the place of the engineer in our modern civilization the purpose has been not to exalt but rather to bring into emphasis the opportunity for service as the measure of the responsibility which attaches to us, both in our individual and collective capacities.

If we, as a guild, stand as the repository of the results of the work of our fellows in bygone ages, even from the prehistoric days

down to the present time, and as trustees for its useful application to the requirements of our own times, then must we, as a guild, recognize our responsibilities to worthily carry on with this great accumulation which has been placed in our hands. It is only the simple truth to say that the trend of civilization in coming ages — especially as to its material content — will depend in fundamental degree upon the manner in which we discharge our obligations during the short day in which this responsibility and opportunity are placed in our hands. We pass this way but once; the opportunity will not come again. Future ages and future generations are waiting the enjoyment of a *material* civilization, the character of which is, in some measure at least, in our hands to determine, here and today.

How the Engineer May Best Serve the Cause of Advancing Civilization

If now we turn to more practical aspects of the matter and ask how we may so act as to properly discharge these duties and responsibilities, we come fairly face to face with the simple query: What is the duty of the engineer of today, and how may he best serve the cause of advancing civilization?

The discharge of a duty is, in large degree, a personal matter; we cannot easily lay down directions for the individual, but, in the aggregate, we can perhaps safely venture to indicate what seems to be the line of obligation.

In the first place, it is perfectly clear that each individual owes it as a duty to himself, to his guild, and to the world at large that he cultivate to the highest practicable degree the faculties and gifts of which by heredity and environment he has become the living expression.

This is simply an application of the parable of the talents, and there is no reason for argument or discussion. Its application and the resultant obligation upon the individual are self-evident.

But individual bricks do not make a building, nor isolated human units a society. If there is any one lesson to be drawn from a study of the evolution of our civilization, it is that of the significance of coöperation. This is the mortar which serves to bind together the individual units and make of them a coherent, enduring, and purposeful structure.

And so, binding together the individual units of our society, there must be the spirit of coöperation; and if this is true for society at large, it is doubly so for those of the guild of the engineer. No one is better qualified to realize the basic law of mechanics that mass effect can be realized only by mass action. The time has gone by when man might be measurably sufficient unto himself. This is the age of great undertakings, and these can be realized only through joint and coöperative effort.

The Part Engineering Societies Have Played in the Developments Signalizing the Last Half-Century

And so, in order to further such ends we as engineers have organized ourselves into groups with closely allied interests and purposes, and have thus formed the great engineering societies of the present day.

If we form, as best we may, in mental vision some composite picture of these great organizations the world over, we shall obtain a most impressive realization of the application of this principle of coöperation and mutual helpfulness. And no one who will study this picture will fail to realize the significant part which these societies have played in the great engineering developments which have signalized the last half-century. It is not perhaps too much to say that the organization of these societies, furnishing as they have great centers around which engineering activities have crystallized, has constituted the most important single element in making possible these latter-day achievements in engineering; and that during this time they have played a vitally essential part in this march along the lines of material progress.

And here again, what has been said with regard to the development of the individual will furnish a sure guide when applied to these great organizations as groups of individuals.

The engineering societies, collectively and individually, will best serve the public welfare and best contribute to the progress of civilization by developing, each in its own sphere, to the highest possible degree. The picture of one great engineering society embracing all who may call themselves engineers and covering the whole field of engineering activity and service, is indeed a beautiful ideal. It seems hardly practicable, however, having in view the great diversity of interest and character of work, and the limitations of the individual regarding the extent of the field which he himself can cover. Were indeed any such society organized, it would inevitably divide up into sub-groups, divisions, sections, and what not, each one corresponding to some one measurably narrow field of interest and activity, even as we note similar centrifugal tendencies in the older and more widely comprehensive societies of the present day. And so we may conclude that intensive development, each within its own boundaries, and assiduous cultivation, each of its own field of activity, will best contribute to swell the grand total of service which the world expects of these great organizations, and which they are under obligation to render.

But the parallel between the individual and the society must not stop here. If coöperation is necessary between and among individuals in order to achieve larger ends, so also is it necessary between and among societies in order that *they* may achieve the larger ends which the progress of civilization demands. There

are many problems of wide sweep for the solution of which the world is looking to the engineer — or to the engineer in alliance with his brother the scientist — and which touch simultaneously the fields of activity of several of our modern engineering societies. These cannot be adequately and properly studied by any one body or group alone. All the mental acumen which can be brought to bear upon these problems will be none too much to light up the pathway toward some solution. Again, the lines of demarcation between the fields of activity of our several societies are often vague, and many fields of activity may fairly enough be claimed by more than one of them. All of this is of course well known, and is a natural result of the marvelous growth in the activity of the engineer during the past half-century and the vast extension in the scope of his work. This situation, however, should lead only to the exhibition of a spirit of kindly coöperation and friendly emulation, motivated throughout by a readiness to unite wholeheartedly in joint undertakings, whenever or wherever such procedure may indicate a more useful result than through individual society effort.

Some Questions for Engineers in their Individual and Collective Capacities

Let us attempt to gather up in a single sweep of vision some concept of the great guild of the engineer with its historic background, tracing far away to the very first impulses toward civilization and with its unique and important place in the fabric of our present-day life; and with this concept in mental vision, will it be amiss if we ask ourselves a few questions?

Do we, either as individuals or as societies, often enough seek out the mountain tops and endeavor, from such vantage point of view, to place ourselves in relation to the great problems and movements of the day and to properly orient our own purposes and aims in relation thereto?

Are we, both as individuals and as societies, too prone to the microscopic view rather than the telescopic?

Are we so occupied with the immediate task, with that which is set before us as the day's work, that we are in danger of failing to appreciate the articulation of our own task with that of our fellow, or to give thought as to how his task and ours may best fit in to the great problems which the progress of our civilization presents?

Have we, either as individuals or as societies, sufficiently well-defined goals or purposes in our professional life? Have we definite aims professionally toward which we are working with a conscious purpose, or are we following too much the opportunist policy of dealing as best we may with individual problems as they arise, and of doing the day's work as it comes along?

Again, are we, either as individuals or as societies, sufficiently alive to the opportunity and duty of taking constructive and helpful part in the larger life movements about us? Do we sufficiently realize the obligation which lies upon us of contributing, as opportunity may offer, helpfully to the solution of great problems which may seem to lie aside from the normal field of our professional life?

Do we, in short, sufficiently realize the obligation of good citizenship along and parallel with the day's work in our chosen field of activity?

Are we, in brief, both as individuals and as societies, striving to definitely direct our course over the sea of life with a firm hand at the helm, with definite objectives in view, with a long look ahead, with a generous recognition of our duty to the generations coming after, and with an appreciation of the larger duties of life as members of a great social organism? Or, are we looking only at our immediate environment with its professional problems, and allowing ourselves, with reference to the larger aspects of life, to drift, subject to currents and tides of which we take little or no heed?

These are indeed searching questions. You of the profession are as well able to answer them as am I. But it will perhaps be safe to assume that neither as individuals nor as societies is our score in these respects what it should be.

If then we imagine ourselves upon the mountain top, may we spend a few moments in looking a little more closely at some few of the things which we might thus discern, and seeing, ask ourselves whether or not we are living up to the full measure of our obligations thereto.

The Duty of the Engineer as Regards Conservation of Natural Resources

Among the many which might thus challenge our attention, none is perhaps of greater importance than that aspect of the duty of the engineer which centers around the general term "conservation of natural resources."

We have already noted the peculiar position in which the engineer stands relative to the constructive materials of the earth and to the inorganic energies of Nature. He is their custodian, and charged with the duty, and has assumed the responsibility, of their development and use to meet the requirements of human progress.

Now the facts are, as engineers well know, that the supplies under these categories are far from unlimited in extent. We know of this fact of limitation and that in some instances it is defined with relative sharpness. Furthermore we have no assurance of the operation of natural agencies tending toward replacement, at least in any degree commensurate with the rate at which we are carrying on with their exploitation and exhaustion. Future generations may perhaps grow beyond the need of some of the things

which we now find necessary. Substitutes may be found in some cases. But we cannot be sure of this, and in the absence of such assurance it does devolve upon the engineer as a most weighty obligation that he give heed to the means which he employs in the exploitation of these natural resources, and that every possible effort be made to avoid waste and to use them with the highest attainable degree of economy and efficiency. Avoidable waste is a direct theft from generations to be. With the lack of care which has characterized much of the exploitation of the resources of Nature, especially during the last half-century, have we not wasted, in some measure, the patrimony of our children and grandchildren? Imagine that in some way the progress of civilization had been hastened and that the exploitation of the resources of Nature had begun on a grand scale with the Romans, two thousand years ago; and suppose that they and their descendants had carried on, as have we during the last fifty or one hundred years; in what condition should we be today? We are not to infer that the sole responsibility in these matters lies with the engineer — he may fairly share it with society as a whole; but after all, his is perhaps the major share because no one is so well qualified as he to see the consequences of the reckless exploitation of Nature, and upon no one does the duty lie so clearly to lift up his voice in protest and to direct his professional energies and skill along lines looking toward the reduction of waste and inefficiency to the lowest minimum.

To this latter duty he has, in fact, responded, and in many cases with most gratifying results. But the end is not yet, and only by untiring, whole-hearted, continuing effort along such lines can the engineer in anticipation look fearlessly in the eye his brothers of the next generation, or the next century, and await their estimate of the manner in which he has discharged the duties of his own day and generation.

Again, if from our fancied mountain top we look in another direction, we shall see ourselves as legatees of a hundred thousand years and more of a gradually growing accumulation. Each age and generation has added some quotum to this vast accumulation of which we are the trustees and executives. Can we either as individuals or societies take a just sense of pride in what we are accomplishing from year to year in this respect? And coming closer home, are we as a society discharging our whole duty in the matter of research and its support? With our numbers and our resources, should we not take a more aggressive stand in this matter and devote, year by year, a larger share of our energies and of our resources to the support of research along lines which lie within the field which we consider peculiarly our own?

The Renewal of Personnel in the Engineering Profession through Technical Education

But let us turn again and note another aspect of our place in the present fabric of civilization. Our guild is like a great army, constantly in need of new recruits in the ranks of the younger strata as its numbers become depleted by age and casualty in those of the older. This is, in short, the great problem of the renewal of personnel, and translates itself into the problem of education for those who are planning to enter our ranks.

It will probably not be claimed that either as individuals or as societies have we in general shown the degree of interest in this subject of engineering education which its importance requires. The technical schools and colleges are intended to furnish opportunity for the effective training of those of our youth who are to furnish at least a very considerable quota of the new recruits in our ranks. Other things equal, we are more likely to find the leaders of future years among those who have been so trained. Of what vital importance it is, then, that we should coöperate with those whose duty it is to direct these great educational activities, to the end, on the one hand, that their measures may be taken with full knowledge of the ever-changing requirements on the firing line of the profession, and on the other, that those who are to receive these new recruits and direct them into the active work of their calling may do so with a sympathetic knowledge of the scope, character, and necessary limitations of training which can be given in the course of a technical curriculum.

As one with some years of experience as a teacher, I am only too ready to admit the inadequacy, in some respects, of the training which our engineering and technical schools are now giving, due in large degree, I believe, to the lack of sufficient contact between the schools and the fields of activity in which their output, as an educational product, is expected to take its place. On the other hand, I believe that some, at least, of the criticism which has been directed toward the product of our technical schools and colleges has arisen as a result of asking or expecting the wrong things of the neophyte — as a result of a lack of understanding or appreciation of the limitations with which the new recruit must approach his job. These conditions can certainly be vastly ameliorated by a better understanding on both sides. The schools and colleges are certainly desirous of such a better understanding, and to us, as societies, the duty now comes, with a significance and an importance which it would be hard to exaggerate, to arouse ourselves to a more active interest in this subject of the training of the recruits for our guild, and to insure that, as far as may be humanly possible, this training shall be such as to best give a well-rounded development of the mental faculties, stimulate genius — if there be such a spark — and withal awaken and foster those characteristics

which will make for accurate thinking, independence, originality, devotion to truth, leadership, and high character.

As we know, a splendid beginning has been made toward this closer approximation of the educator and the requirements of the field of active practice, by way of the survey which is now being carried on through the agency of the Society for the Promotion of Engineering Education.

This is a subject of vital importance to the future of our engineering societies and to the whole future trend of progress along engineering lines. It should receive unqualified and whole-hearted support from these societies, and nothing less will clear us of the obligation which we are under to prepare, as best we may, for the next generation of leadership in our guild.

The Engineering Method and Its Application to Public Questions

The duties and obligations to which I have directed your attention thus far have lain in close relationship to our work as engineers. But there are wider duties and obligations. We are engineers and as such hold a position of peculiar trust and responsibility in connection with the progress of civilization. But we cannot live unto our guild alone. We are citizens of a complex civilization and touch on every hand problems of life and destiny in which we must take some part.

In a sense, life is a complex of problems. The interrogation point presents itself to us on every hand, and in regard to every relation in life.

But the solving of problems in his own field is, in a peculiar sense, the meat and drink of the engineer. His professional work is, in very large part, concerned with just this form of activity, and he has developed and used, consciously or unconsciously, a form of grand strategy which he has found absolutely essential for the effective study of these situations in life.

Thus he knows that a problem presents in general a complex of factors, and that as the first step such factors must be recognized and listed; and furthermore that such a census must be exhaustive — that no factor must be omitted. Again, he knows that such factors must be evaluated in one form or another, that their interactions must be studied — all with a view to their relations to the particular character of the conclusion which it is desired to draw.

And then with all this material in hand it must be subjected to some logical process — formal or informal — and a conclusion drawn. Often, in fact as a rule, the material resulting from the census and evaluation of factors is of necessity incomplete. In many cases the logical process must be informal rather than formal. In all such cases judgment must supply the missing elements if a solution is to be reached. However, no one knows better

than the engineer the need of discrimination between the sure ground of known data and formal logic, on the one hand — as exemplified, say, by mathematical operations — and acts of judgment on the other; and no one has learned through wider experience than the engineer the need of applying his conclusions in the light of that component part which, of necessity, has been dependent on estimate and judgment.

But if such, in broad outline, are the characteristics of the grand strategy which the engineer is accustomed to apply in the study of problems in his own field, how or where could we find a better mode of approach for the study of all problems in life, whether of economics, diplomacy, international affairs, problems of the nation, of the state, of the municipality, of the school district, of trade and commerce, of finance, of education — in a word, of all relations in life which go to make up the complex of our modern civilization?

This does not mean, of course, that the engineer as such can pose as an expert in the study of problems in these varied fields remote from his own normal activity. It does mean that his own general grand strategy is equally applicable in such fields as in his own, and therefore to that extent is he qualified to serve effectively with others who may be able to supply the more narrowly technical details, in the study of a wide variety of problems in life and lying outside his own special field. It means, in particular, when such problems involve questions of engineering or when they have an engineering background, as is so frequently the case, that he is especially well qualified to take an important and helpful part in the broad and thorough study of such matters, and that in general, aside from narrow technicalities, he may helpfully join with his fellows from various walks of life in the effective study of a wide and important range of problems which lie outside the immediate limits of his own chosen field.

It is, in fact, perhaps not too much to say that as the engineering method, if we may so term it, is the more applied in our study of public questions, and broadly in that of the problems of life generally, so shall we be able to reach more sure and safe conclusions, and so will the engineer the more fully realize the degree of service which he may render to the cause of human progress.

Conclusion

To sum up the whole matter, the engineer, either as an individual or as a collective type, is simply a link in the chain of human progress — a chain the links of which, in one form or another, run back into a past removed from our own time by tens of thousands of years, to go to no higher figures. With the trend of human progress as it now is, he seems, moreover, to be a very necessary link. He has taken upon himself the peculiar function of develop-

ing and translating into use for the needs of civilization the constructive materials of the earth and the inorganic energies of nature, and in connection with the exercise of such function he has acquired peculiar and weighty duties and responsibilities.

There are naturally the duties of self-development and improvement, both individually and collectively as organizations such as our own. This is the duty so well inculated by the Scriptural parable of the talents. Likewise there are the duties of friendship and of coöperation for the realization of larger ends, and again, both individually and collectively as organizations.

And then it is peculiarly the duty of the engineer to see that, so far as in him may lie, these stores of Nature, of which he is the custodian, are used frugally, with due regard to their limited supply, and having in mind the needs of future generations. Again, it is his duty to leave behind him some definite increment to that great store of knowledge through which we are able to enter into partnership with Nature, and only by means of which we may hope to more effectively align ourselves with her laws, and thus maintain an ever-ascending gradient of human progress.

Again we must individually as we may, and collectively with definite purpose, endeavor to coöperate helpfully with agencies charged with the training of recruits for our ranks, to the end that there may be a continued and adequate supply to the younger strata in our guild, whence we may hopefully look for leadership and guidance in the future.

And finally, since in the exercise of his functions as an engineer he must of necessity develop and employ habits of mind and methods of study which may be usefully employed in dealing with problems as they arise in all activities in life, therefore should the engineer stand ready to serve, not only in his chosen sphere, but wherever and whenever his habit of mind, his training, and his experience may enable him to contribute a helpful element in this great coöperative enterprise which we call civilization.

No. 1972

ANNUAL REPORT OF THE COUNCIL 1925

THE Council has pleasure in presenting to the membership an outline report of its work and that of the standing administrative and technical committees of the Council covering the administrative year of President William F. Durand.

The personnel of the Council, its officers, and the committees who have served during the same period appear in the Society Affairs section of this volume (see pages 7 to 32). The details of the administrative work of the Council are carried out through standing administrative and technical committees.

It is the endeavor of the President and the Council in the appointments of committees to distribute their membership as widely as possible, to provide rotation in administrative committees, and to distribute committee membership so that the same individuals serve on only one standing administrative committee.

Council meetings have been held in New York, Washington, D. C., Charlotte, N. C., Milwaukee, Wis., and Altoona, Pa.

SPECIAL APPOINTMENTS AND EXCHANGE OF COURTESIES

Representation of the Society has been invited in many different ways in addition to the committee assignments and is taken care of by the appointment of Honorary Vice-Presidents. The Council gratefully acknowledges the service rendered by these members, often at considerable personal sacrifice as there is no appropriation by the Society for such representations. Through 1925 we have been represented as follows:

American Academy of Political and Social Science, Philadelphia, Pa.: H. Birchard Taylor, C. N. Lauer, R. H. Fernald.

American Ceramic Society and Thirtieth Anniversary of the Founding of Collegiate Education in Ceramics, at Ohio State University: Prof. W. T. Magruder, E. F. Du Brul.

American Society for Steel Treating, Cleveland, Ohio: Col. E. C. Peck.

Carnegie Institute of Technology Commencement, Pittsburgh, Pa.: Julian Kennedy.

Congrès International de la Presse Technique, Paris: General Westervelt.

Inter-American High Commission: R. T. Crane, Jr., E. M. Herr, Ralph E. Flanders, E. A. Johnson, Samuel M. Vauclain.

International Congress for Scientific Management, Brussels, Belgium: Chas. de Freminville, Hon. Member.

International Petroleum Congress, Tulsa, Okla.: Homer R. Pierce.

Karlsruhe University, 100th Anniversary, Munich: Prof. Charles D. Marx.

National Rivers and Harbors Congress, Washington, D. C.: Wm. H. Wiley.

Norwegian Engineering Society, 50th Anniversary Meeting, Christiana, Sweden: Otto Kahrs, Peder Lobben.

Pan-American Commercial Congress, Houston, Texas: John Marshall Robertson, James Anderson, Gurnsey A. Palmer, Charles A. Andres.

Regional Planning—American Institute of Architects Conference, New York: J. Roland Bibbins, Elihu Church, Selby Haar.

Rutgers University—Inauguration of John Martin Thomas, President: Roy V. Wright.

Worcester Polytechnic Institute—Inauguration of Captain Ralph Earle, President: Walter M. McFarland.

Sixtieth Anniversary Czechoslovak Society of Technical Engineers, Prague, Czechoslovakia: V. A. Zehr, F. M. Oppermann, Robert Suczek, Vojtech Puc.

Stevens Institute of Technology, Commencement: F. R. Low.

Vanderbilt University Semi-Centennial: H. H. Bailey.

Verein Deutscher Ingenieure and opening of Deutsches Museum, Munich: Dr. Conrad Matschoss.

Committee on Patent Office Procedure: Wallace Clark, W. H. Leffingwell.

Conference American Engineering Standards Committee on Standardization of Overhead Line Materials: Percy H. Thomas.

Conference Department of Commerce to consider adoption of standard inquiry, purchase orders and invoice forms: Frank E. Webner.

Conference of War Department and Representatives of the Construction Industry: President Durand, Secretary Rice, Charles T. Main, Frank A. Scott, John Lyle Harrington.

Joint Conference on Surety Bonds: R. C. Marshall.

Street Highway Conference: E. Posselt.

American Association for the Advancement of Science, Kansas City meeting: Past-President John Lyle Harrington, George C. Shaad.

Unveiling of the monument of General Jose de San Martin, Washington, D. C.: Colonel C. H. Crawford.

Fourth Annual Asphalt Paving Conference: H. H. Esselstyn.

American Society for Municipal Improvements, Des Moines, Iowa: Thos. L. Wilkinson.

The Junior Institution of Engineers of London has been added to the group of societies with whom we exchange courtesies. The spirit of this exchange of courtesies generally covers only the use of the library as a reference room, the benefit of the advice of the secretarial staff, and in some cases the privilege of a mailing address.

SOME COOPERATIVE INTERSOCIETY AND PUBLIC RELATIONS

Engineering Foundation

May 14, 1925, was the tenth anniversary of The Engineering Foundation, and on that date a testimonial dinner was given to its founder, Ambrose Swasey, Past-President and Honorary Member of the A.S.M.E.

National Research Council

The National Research Council, at its meeting of May 18, confirmed the nomination of Earle Buckingham as A.S.M.E. representative on the Engineering Division to fill the vacancy caused by the expiration of the term of office of Albert Kingsbury, serving with Geo. A. Orrok and R. J. S. Pigott.

American Engineering Standards Committee

The A.E.S.C. now numbers about thirty-five member-bodies. It endeavors to act coöperatively as the national clearing house for engineering and for industrial standardization as a federation of the principal organization and governmental departments that are active in such work. It coöperates in international standardization and provides an information service on engineering and industrial standardization.

Its work representing most of the branches of industry continues to grow. A total of 159 projects have been submitted, including sixty-eight approved standards. The record follows:

Projects

Group	Total	Approved
Civil Engineering and Building Trades	32	16
Mechanical	26	10
Electrical	17	5
Automotive	4	1
Transportation	9	6
Shipbuilding	1	..
Ferrous Metallurgy	9	7
Non-Ferrous Metallurgy	14	6
Chemical	12	9
Textile	2	1
Mining	16	1
Wood	5	2
Pulp and Paper	1	..
Miscellaneous	11	4

The A.S.M.E. is a member-body of A.E.S.C., and as such in sectional committee work is responsible in the above list for the development of four standards and joint sponsors on thirteen.

The A.E.S.C. Year Book published in the spring of each year gives complete lists of the projects, and its membership, both member-bodies and sustaining members.

International Electrotechnical Commission

The Council at its meeting of March 16, 1925, approved our representation in the International Electrotechnical Commission and appointed Dr. W. F. Durand and Dr. F. R. Low as our representatives on the U. S. National Committee.

Annual Tables of Constants and Numerical Data

The Society contributes annually to assist in the publication of these tables under the authority of the International Union of Pure and Applied Chemistry through an international commission created for the purpose. The yearly volumes contain all the numerical data—physical, chemical, engineering, metallurgical, biological, etc. These data are collected in the various countries of the world by national committees of abstractors.

International Critical Tables

The Society made a special contribution of $125 toward the publication of the first five volumes of the International Critical Tables of Numerical Data of Physics, Chemistry, and Technology. The values of the various constants and properties which they are to contain are those selected by experts after a critical examination of all data available in the literature. The Annual Tables and the Critical Tables supplement each other, the first being data collected during the periods

between revisions of Critical Tables and serving as the raw material for the experts who prepare the latter.

The publication of International Critical Tables is supported by funds derived principally from industrial organizations. They are issued under the authority of the International Union of Pure and Applied Chemistry and the International Research Council. The financial and editorial responsibility lies entirely in American hands and the Tables will be published in the United States with English as the principal language.

Engineering Education

The A.S.M.E. is one of the coöperating societies with representatives on the Board of Investigation and Coördination of the Society for the Promotion of Engineering Education working under a grant from the Carnegie Corporation, making a comprehensive investigation or survey of engineering education.

The preliminary reports cover the fact-finding of the investigation. The central problem of the investigation is the determination of the fundamental purposes and functions of the engineering colleges in the fields of education, industry, professional life, and public service.

As our part, for mechanical engineering, a questionnaire has been sent out to determine what in general should be emphasized in the mechanical engineering schools curricula.

The report of our A.S.M.E. representatives on the board was presented at the 1925 Annual Meeting. Our representatives are Past-President John Lyle Harrington, F. A. Scott, member of the Council, and W. L. Durand, acting as alternate for Mr. Harrington and Mr. Scott.

Washington Award, Western Society of Engineers

On the Board of Washington Award of the Western Society of Engineers, the A.S.M.E. is represented by two members of the society appointed by the Council. James Lyman was the appointee this year, serving with Dr. Chas. Russ Richards.

The award was made to J. Waldo Smith, member of the A.S.M.E. "for his rare combination of vision, technical skill, administrative ability, and courageous leadership in engineering."

Employment Service

The coöperative Bureau of Employment Service under the joint management of the four national engineering societies is now maintained by contributions from the societies and the members directly benefited.

A Chicago office has been opened at 53 West Jackson Boulevard and is located in the same suite as the offices of the Western Society of Engineers, but separate from their rooms.

A San Francisco office has also been completely organized with offices in the Mechanics Institute Building, which also houses the San Francisco Engineers Club.

United Engineering Society

The Annual Report of the United Engineering Society is made in December of each year covering for the calendar year a summary of operation of the building, operation of the library, and report on funds and property of which the United Engineering Society is the holding corporation and in all of which this Society has one-fourth interest and responsibility.

John Fritz Medal

The twenty-first award of the John Fritz Gold Medal has been made to John F. Stevens, of New York, for "great achievements as a civil engineer, particularly in planning and organizing for the construction of the Panama Canal; as a builder of railroads, and as administrator of the Chinese Eastern Railway." The A.S.M.E. is represented on the Board by four members.

National Museum of Engineering and Industry, and Museum of the Peaceful Arts in the City of New York

The founder societies have been invited to join in the project of a National Museum of Engineering and Industry, and initial steps have been taken for a complete organization. Samuel Insull heads the movement as president. The A.S.M.E. is represented by Fred R. Low and L. P. Alford.

Announcement was made of a provisional endowment of $3,500,000 in the will of our Past-President and Honorary Member, Henry R. Towne, for a Museum of the Peaceful Arts, in the City of New York.

Members of the Society are directors in both museum enterprises, which will undoubtedly be successfully coördinated.

Library

The library of United Engineering Society is jointly administered by a library board composed of representatives from the four founder societies. A complete report of the work of the library is issued in February of each year.

War Construction Engineers Vindicated

The A.S.M.E. was one of many engineering societies who through resolutions urged the quashing of the indictment made in error by the Government against several prominent and long-honored members of the engineering profession, who were members of the Committee on Emergency Construction of the War Industries Board, and who were indicted in December, 1922, "for conspiracy to defraud the Government through control of the construction program during the late war." The charges which challenged their integrity and honesty of purpose were known to be false by men who were in a position to judge, but nevertheless the shadow of an unjust suspicion was cast.

Complete vindication came late in January, 1925, to the seven members of this committee: Benedict Crowell, Cleveland, former Assistant Secretary of War; William A. Starrett, New York; C. W. Lundoff, Cleveland; Morton C. Tuttle, Boston; John H. McGibbons, Chicago; Clair Foster, New York; and James A. Mears, New York.

AMERICAN ENGINEERING COUNCIL

Honorable James Hartness, Past-President of the A.S.M.E., was elected president of the American Engineering Council in January, 1924. The Society had eighteen representatives on the Council and five on the Administrative Board, with President Durand as chairman of the A.S.M.E. delegation.

The major activities of the A.E.C. for 1925 cover the following projects:

Inventory of Water Resources

The Water Power Committee drafted a bill which provides for increased appropriations for the Water Resources Branch of the U. S. Geological Survey. The bill provides for a twenty-year intensive program. An unusually favorable reaction followed the announcement of this proposed legislation, and our local sections did some splendid work in its support.

Committee on Patent Office Procedure

This committee was appointed in the summer of 1924 by Secretary Work to make a thorough study of the Patent Office. When the Patent Office was transferred to the Department of Commerce in April, Secretary Hoover requested the committee to continue its work, adding Messrs. Wallace Clark and W. H. Leffingwell, members of the A.S.M.E., who with the Executive Secretary of American Engineering Council compose a sub-committee for survey of the organization and internal operation of the Patent Office.

Street and Highway Safety

The A.E.C. committee on Street and Highway Safety requested all technical societies to discuss the subject of street and highway safety, and to report any important information or suggestion developed at such meetings. The response was splendid. It is the purpose of the committee to formulate from such information a report to the Second Conference on Street and Highway Safety.

City Building

The Administrative Board authorized the appointment of a Committee on City Building, to encourage and assist local societies to form like committees for the purpose of furthering city planning in their respective communities, and to accumulate information for the use of local committees.

Technical Positions in Public Service

In several localities of the United States public positions requiring engineering or technical training, skill, and experience are being filled by men not possessing such qualifications. Since this practice is inimical to the public welfare, the Board passed a resolution recommending that public positions requiring engineering and technical training be filled only by qualified persons and that all state and local engineering and allied technical societies be urged to coöperate with the public to obtain such trained public officials.

Topographical Mapping

With the President's signature, the Temple Topographical Mapping Bill became a law on February 27. For over three years American Engineering Council has worked to secure the enactment of this legislation. Its final success is due in a large measure to the earnest and effective work which the member-organizations have done in the various states.

The United States Geological Survey is authority for the statement that the exhaustive topographical work will entail the making of 6000 sheets of maps and provide a comprehensive atlas of the United States.

Survey of Air Transportation

The exhaustive survey of air transportation became possible through the appointment by Secretary Hoover of a committee of six to undertake the study, thus securing for the Council governmental assistance and backing. J. Walter Drake, Assistant Secretary of Commerce, directed the work as chairman. A field staff organized with Professor Joseph W. Roe, Head of the Department of Industrial Engineering, New York University, and chairman of the A.E.C. Committee which submitted the preliminary report, as Director and Vice-Chairman of the committee. The other members of the committee are: Dr. W. F. Durand; Professor E. P. Warner; Luther K. Bell, of Washington; traffic manager of the United States Air Mail; C. T. Ludington.

The A.E.C. is also at work on the subjects of national importance as represented in an advocacy of a Division of Public Works, Committees on Reclamation, Reforestation, and Timber Supply, National Board of Jurisdictional Awards, Compensation of the Engineer, et al.

ABSTRACTS OF COMMITTEE REPORTS

Finances

The report of the Finance Committee follows:

The financial condition of the Society on September 30, 1925, may be understood from the balance sheet and statement of income and expenditure as submitted by the auditors, a copy of which forms an appendix to this record.

1924-25 was a notable year in the financial life of the Society. The increase in members' dues went into effect on the first day of the fiscal year. By means of the increased income as well as by a rigid adherence to the budget, the invested reserve of the Society was increased by $68,000. The business affairs of the Society were well conducted, as is evidenced by the successful collection of the dues at the new rate and the small amount of accounts receivable carried over at the end of the year.

While the increase in dues increased the Society's income approximately sixty thousand dollars, its influence on resignations and suspensions cannot be exactly measured, for during the past eighteen months the Society has also exacted a more regid requirement for the prompt payment of dues which has had the effect of increasing the number suspended. Table 1 shows the changes in membership for the past five years.

At the recommendation of the Finance Committee, Council approved the retention of a firm of accountants, Ernst & Ernst, to make a survey of the organization and methods of the Society, in order to check up through competent outside sources on the policies and methods in vogue in the Society's administration. On July 20, 1925, they submitted a report with their comments and recommendations covering organization, balance sheet, statement of income and expenditures, accounting, and publications. The report is under consideration of the Council with the comments of the Finance Committee.

Membership

Table 2 shows the changes in the membership of the Society during the fiscal year 1924-25.

The applicants for membership are published in each issue of the *A.S.M.E. News*, and balloted on each month by Council.

The Council has the pleasure of recording the election to Honorary Membership in the Society this year, of:

TABLE 1 CHANGES IN MEMBERSHIP, 1920–1925

Fiscal year	Total members	Applications received	New members		Resignations		Suspended		Died		Net increase	Decrease
			No.	Per cent	No.	Per cent	No.	Per cent	No.	Per cent		
1920–1	15015	2712	2468	16.8	116	0.8	501	3.3	87	0.6	1764	....
1921–2	16696	2229	2071	12.4	146	0.9	151	0.9	92	0.6	1681	....
1922–3	17938	1915	1647	9.2	127	0.7	149	0.8	74	0.4	1297	....
1923–4	17594	2152	1841	10.5	352	2.0	1730	9.8	103	0.6	....	344
1924–5	17668	2249	1475	8.4	577	3.3	779	4.4	119	0.7	74	....

TABLE 2 MEMBERSHIP CHANGES, OCT. 1, 1924, TO SEPT. 30, 1925

	Membership		Losses				Additions		Totals		
	Oct. 1, 1924	Oct. 1, 1925	Transferred from	Resigned	Dropped	Died	Transferred to	Elected	Loss	Gain	Change
Honorary Members	19	21	....	....	....	1	....	3	1	3	2+
Life Members	77	78	....	....	....	2	....	3	2	3	1+
Members	8019	8065	....	185	152	91	174	300	428	474	46+
Associates	863	792	4	69	35	4	5	36	112	41	71—
Associate-Members	4147	4182	78	136	215	13	140	337	442	477	35+
Juniors (20)	1010	1016	116	104	51	2	279	....	273	279	6+
Juniors (10)	3519	3574	400	83	326	6	....	870	815	870	55+
TOTALS	17594	17668	598	577	779	119	598	1549	2073	2147	74+

Worcester Reed Warner, Past-President, in recognition of the high service he has rendered in improving the design and workmanship of astronomical instruments of extreme precision.

William Wallace Atterbury, in recognition of the high service he has rendered in the field of railroad transportation.

Herbert Clark Hoover, in recognition of the high service he has rendered to the profession of engineering, to his country, and to the world.

Memorial Notes of members whom the Society has lost by death during the year have appeared in the *A.S.M.E. News* and in *Mechanical Engineering*, and are recorded in this volume.

The Council records with deep regret the death of Major William H. Wiley, Treasurer of the Society since 1883 and one of its organization members.

The Committee on Membership held eighteen meetings during the year 1924-25.

The number of applications considered in the transaction of its work, and a summary showing the action taken follow:

Applications pending Oct. 1, 1924	409	
Applications received during fiscal year 1924-25	2239	
Total applications handled during year 1924-25		2648
Recommended for membership	1864	
Promotions denied	7	
Deferred	16	
Indefinitely deferred	4	
Withdrawn	61	
Applications pending, Oct. 1, 1925	696	
Total		2648

Those recommended for membership were divided into the following grades:

Members	296	
Promotions to Member	198	
Associates	28	
Transfers to Associate	5	
Associate-Members	289	
Promotions to Associate-Member	190	
Juniors	399	
Juniors (R 5 Rule 1)	459	
Total		1864
Recommended elections declared void	88	
Recommended resignations accepted	271	

Sixty-five Junior members reached their thirtieth birthday during the fiscal year 1924-25 and their status is as follows:

Transferred to higher grades	17	
Applications for transfer pending	11	
Memberships to cease	37	
Total		65
Applications recommended	1864	
Promotions and transfers	393	
Total new Members		1471
Resignations and declared void	359	
Junior memberships ceased	37	
Net gain in Members		1075

Meetings and Program

During the year three meetings of the Society were held. The 1925 Annual Meeting was one of the largest ever held, with a registration of 2027, and the Milwaukee Meeting featured a fine technical program which brought out thorough and well-considered discussion. Both of these meetings are reported in the Society Affairs section of this volume.

The Regional Meeting at Portland, Oregon, brought together a splendid attendance from the Pacific Coast for an interesting meeting which lasted four days.

Publications

The year 1924-25 marked an end of the restricted policy made necessary by the curtailment of expenditures, and the resumption of the publication basis of 1920. TRANSACTIONS was returned to its former size, nearly 1400 pages appearing in volume 46 which was issued in July. The total pages of *Mechanical Engineering* was forty-six per cent greater than during the preceding fiscal year. This includes the special Mid-November number and the supplement to the May issue which presented papers for the Annual and Spring Meetings of the Society in ample time for members to study them carefully before the meetings. The publication of the *A.S.M.E. News*, the 1925 *Year Book*, the 1924 *Engineering Index Annual*, and *Condensed Catalogues* were carried on in the usual successful manner.

Professional Divisions

The Professional Divisions have contributed major portions of the programs for the Annual and Spring Meetings. Members of the Society to the number of 279 are serving on the 62 committees and subcommittees carrying on the work of the 13 Professional Divisions on Aeronautics, Fuels, Machine Shop Practice, Management, Materials Handling, National Defense, Oil and Gas Power, Petroleum, Power, Printing Machinery, Railroad, Textile, Wood Industries.

During the year the organization of the Petroleum Division was completed.

Satisfactory progress is being made to place the special knowledge and experience of the Professional Divisions at the service of the various Local Sections. Nineteen Local Sections now have direct contacts with the individual Divisions.

The Professional Divisions represent an important function for recommending projects for the research, standardization, and safety-code activities of the Society. The procedure to accomplish this is now in process of development. The first step taken at the 1925 Annual Meeting was the presentation, for public discussion, of reports of progress in the various fields of mechanical engineering.

During the year the Professional Divisions have coöperated in many activities with outside organizations. Through the Oil and Gas Power Division a highly successful program for Oil and Gas Power Week was developed in which 18 national bodies and about 100 local bodies participated in 140 meetings throughout the country. Through the Management Division the Society assisted in the conduct of Management Week. The Machine Shop Practice Division contributed an excellent technical program at the New Haven Machine Tool Exhibit, and also at the convention of the American Society for Steel Treating at Cleveland.

Local Sections

The total number of Sections is now 66; 442 meetings were held in these centers.

Two Sections were formed during the year, one in the State of Florida and the other in the State of West Virginia.

An outstanding achievement was the successful conduct of the Fifth Annual Machine Tool Exhibition under the auspices of the New Haven Section at Yale University, September 8-11. The number of visitors exceeded fifteen thousand, and there was a splendid attendance at the technical sessions.

Special mention should be made of the importance of the annual conferences of Local Sections Delegates at the Annual and Spring Meetings.

During the year Local Sections have wholeheartedly supported the fundamental activities of the Society as evidenced by their coöperation in Management Week, in Oil and Gas Power Week, and in the procurement of an increasing number of papers of the standard to warrant publication in *Mechanical Engineering* and TRANSACTIONS.

To facilitate the conduct of Local Sections the Committee has issued a manual for the operation of a Local Section.

Awards and Prizes

The Council has established the following policy:

Awards are made by the Society on the recommendation of the Committee on Awards. Prizes may be established by the Professional Divisions, Local Sections, or other agencies within the Society.

It is not obligatory on the Council to make awards within any one year if the character of the papers or inventions considered does not warrant this recognition.

Charles T. Main Award: The purpose of this award is to encourage young engineers to learn to write good English and to promote a broader study of engineering as an influence to a better social life. The subject for the 1925 award was The Influence of the Locomotive upon the Unity of Our Country, and the winner was Clement R. Brown, of the Catholic University, Washington, D. C.

Student Awards: The awards by Council for student papers were made as follows:

1. Harry Pease Cox, Jr., of Rensselaer Polytechnic Institute, for his paper on A Study of the Effect of End Shape on the Towing Resistance of a Barge Model.

2. Wm. S. Montgomery, Jr., and E. Ray Enders, Jr., of Pennsylvania State College, for their paper on Some Attempts to Measure the Drawing Properties of Metals.

Junior Prizes: Award was made to Gilbert S. Schaller for his Investigation of Seattle as a Location for a Synthetic Foundry Industry.

Gift for Student Aid

The Society has been the grateful recipient of a gift of $15,000 through the generosity of Major Max Toltz, member, endowing a fund to be known as the Major Max Toltz Fund for assistance to students.

Student Branches

The administrative work in connection with the Student Branches is conducted by the Committee on Relations with Colleges. During the year new Student Branches were established at the University of Pennsylvania, the Post-Graduate School at the United States Naval Academy at Annapolis, and at the College of Engineering at the Newark Technical School.

Student Branches held 552 meetings, several of these jointly with adjacent Local Sections. At a meeting of this kind in New York there was an attendance of over 600 students from nine Student Branches. Thirty-five per cent of the members of Student Branches in the 1924 graduating classes have applied for Junior membership in the Society.

The Student Branches were visited to a greater extent this year than before by President Durand, Geo. A. Orrok, Charles Penrose, Chairman of the Committee on Local Sections, and by members of the Committee and the staff.

Student Branch conferences were held at the Annual and Spring Meetings and many excellent ideas developed. Mimeographed bulletins to the various Student Branches were inaugurated during the year.

The committee is anxious to secure the cooperation of members who will be willing to visit the Student Branches and tell them of the work of the Society and of the profession.

Education and Training for Industries

The Committee on Education and Training for the Industries directed its major efforts to the conduct of sessions at the Annual and Spring Meetings of the Society in the belief that public discussion of the outstanding needs for better methods of training in the industries is the best contribution the committee can make to the cause.

Excellent sessions were held at both meetings, with good papers and copious discussion. The papers have appeared either in full or abstract in *Mechanical Engineering*.

TECHNICAL COMMITTEES

Boiler Code

During 1925 work was completed on the Rules for the Care of Steam Boilers and other Pressure Vessels in Service to form Section 7 of the A.S.M.E. Boiler Construction Code.

An important change in the committee's procedure adopted during the current year deals with revisions and addenda to the various codes. Instead of the revision of the printed codes as has been the custom previously, the committee now plans to issue as addenda the various changes that are recommended from time to time.

The committee reports with great regret the resignation of Chairman John A. Stevens, who, since the origin of the committee in 1911, has led the work of the committee in an inspiring, helpful, and painstaking manner.

Standardization

Standardization in the mechanical industries of special interest to the members of the A.S.M.E. has been carried forward. Fourteen sectional committees, including thirty-three sub-committees, have been at work, and some have completed substantial parts of the projects assigned to them. Brief summaries of the progress reports of some of these committees follow.

Shafting: In December, 1924, the A.E.S.C. announced its approval as a Tentative American Standard of the Diameters of Lengths of Shafting, the first report prepared by this committee. After approval of the Standard for Square and Flat Stock Keys by the A.S.M.E. Council in March, 1925, this report was presented to the A.E.S.C. and received the designation of Tentative American Standard in June, 1925.

Plain Limit Gages for General Engineering Work: The first report of this sectional committee, entitled Tolerances, Allowances and Gages

for Metal Fits, was submitted to the Council in March, 1925, and in April it received the approval of the A.E.S.C. with the designation Tentative American Standard. The second report, to be known as Methods of Gaging and Specifications for Plain Limit Gages, is ready to be placed in type for distribution with letter ballot to the members of the Committee.

Pipe Flanges and Flanged Fittings: While this committee cannot point to a report published during the past year, all four of its subcommittees have been actively at work on their tasks and much has been accomplished.

Pressure Piping: For some time conferences and correspondence with members indicated the growing need for a national code on pressure piping. The Standardization Committee accordingly arranged for an informal conference during the 1925 Spring Meeting of the Society for the purpose of obtaining opinions. This conference was unanimous in the opinion that there is an urgent demand for a code on pressure piping. The Council at its October meeting signified its approval of the recommendation that a committee should be organized under the procedure of the A.E.S.C., and the services of this Society as sponsor or joint sponsor have been offered the A.E.S.C.

Standards for Graphics: The Council at its May meeting requested the authorization of a new A.E.S.C. sectional committee on this important subject, and offered the services of the A.S.M.E. as sponsor or joint sponsor.

Drawings and Drafting-Room Practices: The A.S.M.E. Standardization Committee recommended to the Council that preliminary steps be taken to initiate a project of standardizing drawings. The Council approved the proposal in May, and in June the A.E.S.C. was formally requested to authorize the organization of a sectional committee for this purpose.

T-Slots: A questionnaire on the standardization of T-slots was prepared and mailed to approximately 250 firms during the month of October, 1924. The replies were analyzed and at the May meeting of the sub-committee a revised report was adopted. This new draft was further considered at the December, 1925, meeting.

National (American) Standard Fire-Hose-Coupling Screw Thread: In its revised form this standard was approved as an A.E.S.C. American Standard in April, 1925. It, together with the two supporting pamphlets dealing with the gaging of these coupling threads, is now available for general distribution.

Bolt, Nut, and Rivet Proportions: The proposed standard for plow bolts was put to letter ballot of the Sectional Committee in August. The report of Sub-Committee No. 2 on Wrench Head Bolts and Nuts has undergone two complete revisions during the year, the last of which will be considered by the Sectional Committee at its meeting in December.

Scientific and Engineering Symbols and Abbreviations: This Sectional Committee, which is sanctioned under the procedure of the A.E.S.C., completed its organization during the past year.

Research

Two new special Research Committees were organized during the year on Corrosion of Condenser Tubes and Boiler-Furnace Refractories. The Society also joined the Research on Boiler Feedwater by appointing seven representatives on a joint committee.

The Committee on Lubrication is pursuing its study of viscosity under high pressure at the Bureau of Mines Experiment Station, Pittsburgh.

The Committee on Fluid Meters issued Part 1 of its Report in November, 1924.

The special machine designed by the Committee on the Strength of Gear Teeth was placed in the Engineering Laboratory of Massachusetts Institute of Technology. A few months have been devoted to tuning up the machine and making minor changes.

The Committee on Properties of Steam and Extension of the Steam Tables is pursuing its program of research at Harvard University, Massachusetts Institute of Technology, and the Bureau of Standards. Progress was reported at a special session of the 1925 Annual Meeting.

B. H. Blood resigned from the chairmanship of the Committee on Cutting and Forming of Metals. Professor James A. Hall was elected in his place. Professor O. W. Boston, of the University of Michigan, will act as secretary. The committee is at work on its program for research in this field.

The Committee on Metal Springs has set up an aggressive program of research, which has been reflected in excellent sessions at the Annual and Spring Meetings.

The Committee on the Effect of Temperature on the Properties of Metals, a joint project with the A.S.T.M., has held several meetings and is engaged in the formulation of its program.

Safety

During the year, the Council approved the change in name of the standing technical Committee on Safety Codes to the Committee on Safety. This change is accompanied by a broader function than the former mere suggestion and supervision of safety-code projects.

The outstanding accomplishment of the year is the publication in July, 1925, of the first revision of the Safety Code for Elevators, in the formulation of which the A.S.M.E. served as joint sponsor with the Bureau of Standards and the American Institute of Architects.

Permanent officers were elected at the meeting of the Sectional Committee on Machinery for Compressing Air at a meeting which was held on December 4, 1924. Since then some progress has been made toward the formulation of this safety code.

At the request of the officers of the National Bureau of Casualty and Surety Underwriters, joint sponsors with the A.S.M.E. for a Safety Code for Conveyors and Conveying Machinery, the A.S.M.E. committee undertook the organization of the Sectional Committee under the procedure of the A.E.S.C.

Two sponsorships for safety codes under the procedure of the American Engineering Standards Committee have been accepted by the Society during the past year. These include sole sponsorship for a new Safety Code for Cranes, Derricks, and Hoists, and joint sponsorship with the National Board of Casualty and Surety Underwriters for the Safety Code for Mechanical Power Control.

In addition to the committees already mentioned, the A.S.M.E. is officially represented on nineteen safety committees organized under the procedure of the American Engineering Standards Committee.

Power Test Codes

Ten of the nineteen test codes are now in final pamphlet form, five having appeared during the year. Six additional codes will soon be presented for Council approval. Of the remaining codes, the one for Gas Producers is being revised by the committee, that for Steam Turbines was presented for public hearing at the 1925 Annual Meeting, and the Code on Instruments and Apparatus has had several chapters prepared and issued for criticism.

Through the Main Committee on Power Test Codes, the Society is coöperating with the International Electrotechnical Commission in the consideration of international test codes for prime movers.

Society Program

The Council appointed a committee on Society Program consisting of Past-President Dexter S. Kimball, Chairman, Roy V. Wright, Vice-President, and H. V. Coes, Chairman of the Finance Committee. It is felt that the functioning of the Society as a whole can be improved by a program for definite achievement. Preliminary studies have already developed the advisability of fixing a future policy and program in our relations to industry, research, technical discussion, standardization, and safety problems.

APPENDIX

REPORT OF ACCOUNTANTS

Wm. J. Struss & Co., certified public accountants, give the results of their examination of the books of the Society for the fiscal year ended September 30, 1925, in the statement of assets and liabilities on pages 584 and 585.

STATEMENT OF FINANCIAL CONDITION, SEPTEMBER 30, 1925

ASSETS

Current			
Cash on hand and in banks (for Working Capital)	$24,911.30		
Cash on hand and in banks (for Contributions)	4,629.97		
Cash on hand and in banks (for Advance Payments)	1,880.08		
TOTAL CASH		$31,421.35	
Accounts Receivable for Major Liabilities	$30,047.71		
Accounts Receivable for Working Capital	103,887.49		
TOTAL ACCOUNTS RECEIVABLE		133,935.20	
Inventory of Publications for Sale	$17,638.02		
Inventory of Supplies (not merchandisable)	24,122.41		
TOTAL INVENTORY		41,760.43	
Deferred Charges,			
Compiling Publications for Future Sale		15,736.16	
TOTAL CURRENT ASSETS			$222,853.14
Investments for Trust Funds (per contra)			
Lawyers Mortgage 5½—1927	$40,000.00		
St. Louis, Peoria & N. W. 5—1948	10,613.89		
N. Y. Central & Hudson Riv. R. R. 4—1942	23,062.50		
Cash in Banks (to be invested)	2,172.81		
			75,849.20

LIABILITIES

Current			
Unfilled Obligations		$30,047.71	
Contributions (per contra)			
Research on Lubrication	$400.00		
Taylor Society	15.00		
Gear Research Fund	1,391.14		
Standardization Bolts, Nuts, Rivets	526.39		
Fluid Meters Research	1,875.00		
Steam Table Research	122.44		
Metal Spring Research	200.00		
Oil and Gas Power	100.00		
		4,629.97	
Dues Paid in Advance (per contra)		1,880.08	
TOTAL CURRENT LIABILITIES			$36,557.76
Trust Funds (per contra)			
Life Membership Fund		$49,486.06	
Library Development Funds		5,302.77	
Weeks Legacy		2,116.69	
Melville Fund		1,439.07	
Charles T. Main Fund		2,839.20	
Hunt Memorial		275.01	
Hess Prize—Juniors and Students		2,000.00	
Easby Trust Funds		270.40	
Westinghouse Bust and Trust Fund		6,080.00	
Holly Medal Fund		6,040.00	
TOTAL LIABILITIES (Trust Funds)			75,849.20

Investments for Capital Account (Reserve)			
Lawyers Mortgage Bonds 5½—1928-1929-1930.....		134,000.00	
Fixed Assets			
Building and Equipment (Book Value).....	$498,149.09		
Library Books (Book Value).....	13,000.00		
Engineering Index (Book Value).....	10,000.00		
		521,149.09	
			Capital
			Invested in Building and Equipment.... $498,149.09
			Invested in Library Books..... 13,000.00
			Invested in Engineering Index..... 10,000.00
			$521,149.09
			For Reserve Lawyers Mortgage Bonds... 134,000.00
			Working Capital
			Inventory $41,760.43
			Cash 24,911.30
			Deferred Charges 15,736.16
			Accounts Receivable 103,887.49
			186,295.38
			TOTAL CAPITAL 841,444.47
TOTAL ASSETS		$953,851.43	$953,851.43

Total Current Assets.....	$222,853.14
Total Current Liabilities.....	36,557.76
Net Current Assets, Represented by Working Capital...	$186,296.38

No. 1973

NORMAL PITCH—THE INDEX OF GEAR PERFORMANCE

By G. M. Eaton,[1] East Pittsburgh, Pa.
Member of the Society

The paper brings out certain departures from previously accepted practice which are useful in the manufacture of heavy involute gearing, as they ease the performance during the breaking-in stage of operation. It shows that material improvement in performance may be secured by adopting the proper relation between the normal pitches of the driving and driven gears, measured at the point of tooth engagement. Finally, it outlines the development of normal-pitch indicators.

Development and Application of Inspection Methods

THE methods outlined were developed and are in commercial use at the R. D. Nuttall Company. They apply particularly to gears in heavy duty, as, for example, in heavy-traction electric locomotives. The fundamental principles, however, apply to any involute gearing.

2 During the development stage, it was found necessary to adopt close limits for many of the variables involved; but the ultimate aim throughout has been the establishment of the widest limits that will insure acceptable performance. This is being approached by exploration of the effects of various departures from precise accuracy.

Definition of Quiet Gearing

3 The term "quiet" is used here as the opposite of both vibration and noise. Quiet operation is a relative term. All gears produce vibration and noise when transmitting power. Quiet gearing may therefore be defined as gearing which operates with an acceptable degree of vibration and noise. Thus it is possible that gearing which is quiet in one service may be regarded as prohibitive under more drastic requirements.

[1] Chief Mechanical Engineer, Westinghouse Elec. & Mfg. Co.

Contributed by the Machine Shop Practice Division and presented at the Annual Meeting, New York, November 30 to December 4, 1925, of The American Society of Mechanical Engineers.

Class of Gearing under Discussion

4 The gears used for the basis of this discussion are $1\frac{3}{4}$ diametral pitch gears with a nominal one-two tooth contact. They were hob cut and given a drastic heat treatment. In some cases they were put into service in this condition, and in other cases the tooth contours were ground after the gears were hardened. During the development stage, the fine variations of normal pitch discussed in this paper are practical only when at least one of the mating gears has ground tooth contours.

Variables

5 Quiet operation can be assured only by holding a large number of variables within proper limits. These variables group themselves naturally into two main classifications:

a Variables in the gear mounting
b Variables in the gear proper.

6 All the variables in the gear mounting, as, for example, the accuracy of alignment in both planes, the adequacy of bearings, the rigidity of structure, lubrication, critical response to existing cyclic disturbance, and other well-recognized fundamental characteristics, must be held within limits adapted to the degree of quietness that is demanded. In fact, this phase of the problem is mentioned here only to call attention to the ease with which it is possible to ruin the performance of adequate gears by the failure to exercise, in the mounting, a degree of care appropriate to the quality of the gearing.

7 Until very recently it has been customary to tabulate the major gear variables upon which quietness depends as follows, the order of importance dictating their position in the list:

a Pitch (circular or chordal)
b Smoothness of working surfaces
c Contour
d Concentricity.

We wish to bring into the sharpest focus the statement that in this list the most vital fundamentals are only indirectly indicated, and we offer the following as clearly defining the fundamentals of gear teeth on which quiet operation is dependent.

8 There are four major items which inspection must determine:

a Initial engagement of mating teeth (premature, correct, or delayed) as determined by normal pitch. (This is the only one of the fundamentals analyzed in this paper.)
b Concentricity
c Contour
d Smoothness of working surfaces.

9 In setting the limits of normal pitch within which the inspector may accept gears, the main fundamentals which the engineer must analyze are:

a Angular velocity
b Inertia of associated rotating parts
c Elasticity of associated rotating and stationary parts
d Normal tooth pressure
e Lubrication
f External disturbances of cyclic characteristic
g Amplifying tendency of associated structures.

The engineer must also analyze the combination of the first four items of his list as to its tendency toward producing critical response to each of the existing cyclic disturbances.

10 While this sounds very formidable, it must be remembered that quietness is relative, and the degree to which the engineer must pursue the more intricate of these investigations is a direct function of the measure of perfection which he is trying to achieve.

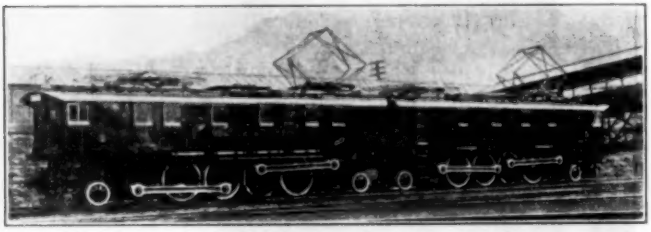


FIG. 1 ELECTRIC LOCOMOTIVE, NORFOLK & WESTERN R. R. CO.

11 The order of importance of the various items may be different for various installations, with the single exception that the character of the initial engagement of mating teeth is probably always the prime function of quiet operation.

SERVICE CONDITIONS

12 The discussion is based upon the main gears of electric locomotives recently supplied to the Norfolk and Western Railroad Company by the Westinghouse Electric and Manufacturing Company and the American Locomotive Works. This installation was chosen because of the exceptional combination of service conditions, rather than because extremely quiet operation was essential. There was, however, a demand that vibration be held within limits that were in keeping with the endurance of associated structures.

13 Fig. 1 shows the complete locomotive, and Fig. 2 the main motor with its overhung pinions, outside of which are mounted collector rings for three-phase current. These collectors and their

associated leads formed the limiting feature which determined the degree of vibration that was acceptable. Fig. 3 shows the main gear, in which tangential, spring-restrained movement is provided between the gear rim and the gear center.

14 When the locomotive is hauling its train up a grade, the pinion drives the gear. This service will be referred to hereafter as "motoring." This service must be performed with either end of the locomotive leading; that is, the pinion drives the gear either clockwise or counterclockwise.

15 When the locomotive with its train is descending a grade, the main motors supply the braking resistance and return energy

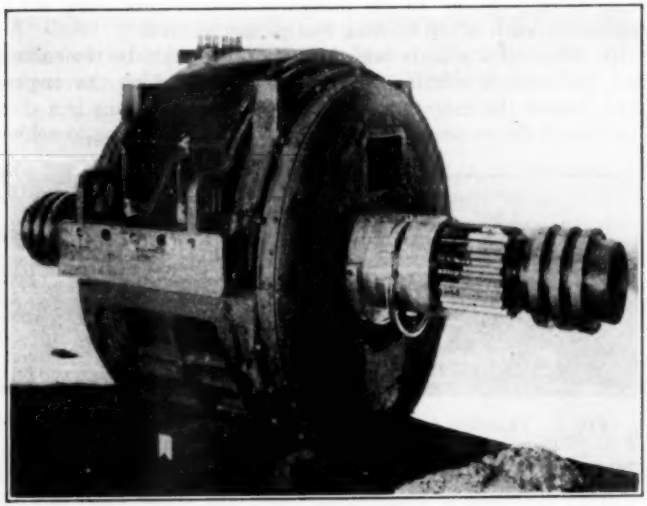


FIG. 2 MOTOR OF NORFOLK & WESTERN ELECTRIC LOCOMOTIVE

to the power line. This service will be referred to hereafter as "regenerating." While regenerating, the gear drives the pinion, and this drive, again, may be either clockwise or the reverse.

16 The connecting rods, shown in Fig. 1, produce an external cyclic disturbance. The discussion will show that quiet performance is harder to achieve in this combination of service than is the case with a simple one-way drive.

ANALYSIS OF TOOTH ENGAGEMENT — FORCES DIRECTED ALONG THE LINE OF ACTION

17 When the mating teeth of a pair of gears engage, forces are set up parallel and perpendicular to the line of action. These forces may be analyzed separately, or they may be combined and the

analysis directed to the resultant. The former method has been adopted because it seems to lend itself better to a clear understanding of the actions which take place. The following phase of the analysis is based upon the assumption that the tooth contour is the closest practicable approximation of the involute form.

18 Referring to Fig. 4, it is evident that just prior to arrival at the indicated position, the entire motor torque was carried at A. When the normal pitch N is equal on the driving and driven gears, the next pair of mating teeth come smoothly into engagement as they enter the line of action at B (neglecting deflections).

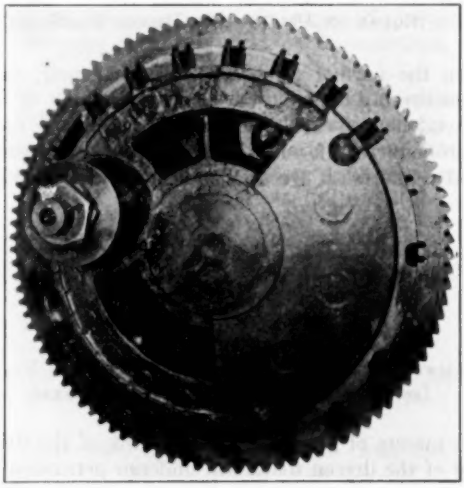


FIG. 3 MAIN GEAR, NORFOLK & WESTERN ELECTRIC LOCOMOTIVE

19 Going a step further, however, we find that deflections interfere with this smooth engagement. Analyzing deflection in detail, we find that it occurs in three ways with new gears:

a The flattening down of high spots

b Surface depression sufficient to develop areas adequate for the support of the existing normal pressure

c Cantilever deflection of working teeth. These deflections, while small, are larger by a material percentage than indicated by the usual cantilever formulas, due to the disturbance of base that occurs when a short, deep cantilever is loaded.

20 The resultant effect of all three deflections is to shorten the active normal pitch of the driving gear, and to lengthen the active normal pitch of the driven gear. Thus the caption under Fig. 4

is true in a precision sense only when rolling the gears without load, if they are made with their normal pitches strictly equal.

21 Fig. 5 shows an exaggerated view of what tends to happen as a result of the deflections noted. Premature engagement occurs at *B*.

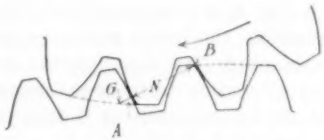


FIG. 4 CORRECT ENGAGEMENT OF MATING TEETH. NORMAL PITCH EQUAL ON DRIVING AND DRIVEN GEARS

22 When the normal pitch of the driven gear, as manufactured, is measurably longer than the normal pitch of the driving gear, the tendency toward premature engagement may become still more pronounced. If, under this condition, the angular velocity is high, and if the teeth are soft and comparatively rigidly asso-

FIG. 5 PREMATURE ENGAGEMENT OF MATING TEETH. NORMAL PITCH LONGER ON DRIVEN THAN ON DRIVING GEAR

ciated with masses of great inertia, the flank of the driving tooth and the tip of the driven tooth will undergo permanent distortion and wear. If, however, the teeth have been sufficiently hardened by heat treatment to enable them to sustain within their elastic limit the forces set up by this dynamic engagement, the positions

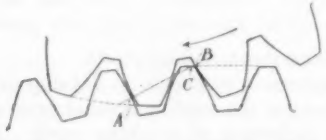


FIG. 6 PREMATURE DISENGAGEMENT OF MATING TEETH. NORMAL PITCH LONGER ON DRIVEN THAN ON DRIVING GEAR

indicated in Fig. 6 will be assumed. The teeth at *A* will break contact as the teeth at *B* pick up the load, thus shortening the length of the line of action to slightly more than one normal pitch. (It will be noted that the line of action is no longer a single straight line, but follows the line *BCA*, Fig. 6. The portion *BC* will be

somewhat modified by local surface deflections, but it is sufficiently accurate to serve as a diagram of what actually occurs.)

23 This readjustment of the relative angular positions of the driving and driven gears can occur in the brief time element allowed, without setting up destructive locally applied forces, only

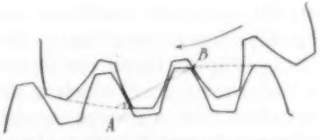


FIG. 7 DELAYED ENGAGEMENT OF MATING TEETH. NORMAL PITCH SHORTER ON DRIVEN THAN ON DRIVING GEAR

by virtue of the elastic characteristics of the entire associated structure. When the elastic conditions, masses, and frequencies, bear to each other a relation permitting resonance to occur, the situation becomes serious. The engineer must therefore predeter-

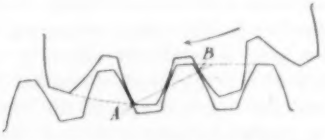


FIG. 8 DELAYED DISENGAGEMENT OF MATING TEETH. NORMAL PITCH SHORTER ON DRIVEN THAN ON DRIVING GEAR

mine as far as possible that the relation of these fundamentals is satisfactory.

24 In certain one-way drives where unsatisfactory operation was recently investigated and corrected, the basic reason was found

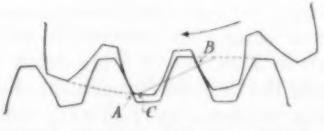


FIG. 9 DELAYED ENGAGEMENT OF MATING TEETH. NORMAL PITCH SHORTER ON DRIVEN THAN ON DRIVING GEAR

to be a normal pitch of the driven gear longer than that of the driving gear.

25 In installations where the pinion drives the gear and little or no back-driving occurs, the sensible course is to avoid dipping into the difficult and somewhat indeterminate investigations of resonant response to premature tooth engagement. This may be

approached by making the normal pitch of the driven gear shorter than that of the driving gear, by an amount sufficient to compensate for the deflection of the active teeth, as shown in Figs. 7 and 8.

26 The limits should be set at values also insuring that premature engagement will be avoided even when the most undesirable extremes of commercial tolerances chance to be cumulative.

27 In the locomotive installation under discussion, however, the issue is more complicated. In all the illustrations the pinion has been shown as the driving gear, which is the motoring condition. But every departure from a precise equality of normal pitch of the gear and pinion which may be adopted to improve motoring performance serves to deteriorate the performance while regenerating.

28 The best overall performance therefore becomes a compromise and involves a review of the relative motoring and regenerating mileages and loadings for which the locomotives are intended. The engineer is thus forced to delve as far as he can into the determination of the forces set up under the conditions of compromise which his judgment leads him to think is advisable.

29 Referring further to the action occurring when the normal pitch of the driven gear is relatively short, it will be seen from Fig. 9 that the teeth at B remain out of contact till the teeth at A have passed beyond the nominal end of the line of action, again departing from a single straight line and following the path diagrammed by the line BCA.

30 Evidently there must be a readjustment of the relative angular positions of the driving and driven gears between the positions of Figs. 8 and 9 before the teeth at B can assume the load. It is obvious that here again there are possibilities of dynamic engagement and resultant resonance which demand investigation. While a close quantitative analysis of the resulting forces is hardly feasible, it is fortunately well within the scope of a simple word analysis to prove that, with identical discrepancies of normal pitch, the dynamic forces arising in connection with the premature engagement shown in Figs. 5 and 6 are much more severe than those accompanying the delayed engagement shown in Fig. 9.

31 The analysis immediately following will show that premature engagement is the exciting cause of severe transient accelerations of the rotating masses, while delayed engagement follows less severe angular accelerations which are produced by the driving torque.

32 The accelerations of premature engagement are directly opposed by the torque of the motor and the moment of inertia of the rotating parts on the driving side of the system, and on the driven side are opposed by the moment of inertia of the gear rim, the friction between the gear rim and the gear center, and the

driving-torque reaction as applied to the gear rim through the radial leaf springs with their inherent frictional damping. It is apparent that the tendency of all inertias, live forces, and frictions is to increase the severity of the dynamic conditions accompanying premature engagement. The only relief offered by the system lies in the elastic characteristics of the entire structure.

33 Some of the conditions surrounding delayed engagement are in marked contrast to those just described. The forces active in producing the angular accelerations required to close the gap at *B*, Fig. 8, and thus reach the positions shown in Fig. 9, are the torque of the motor and the reactions of the radial leaf springs in the gear. The conditions to this point are akin to those of premature engagement, except that cause and effect have changed places. But all inertias and frictions tend to reduce the velocities attained during the closing of the gap at *B*, Fig. 8, and therefore tend to reduce the forces set up at the instant that this gap is closed. Also, the tendency of tooth deflections is to narrow this gap, thus reducing the angle through which acceleration must occur, as contrasted with premature engagement, where tooth deflections increase the space through which accelerations are effective. And finally, under the conditions of delayed engagement, the various masses are not free to respond to the forces enumerated thus far, as the relative velocities of the driving and driven masses are dictated by the continuing contact of the teeth at *A*, Fig. 8.

34 As before, the elasticity of the system cushions the shock attendant upon delayed tooth engagement.

35 From this analysis, it is clear that the relative angular velocity of the driving and driven teeth as they come into delayed engagement, and the resulting impacts, are less than the corresponding occurrences with premature engagement.

Analysis of Tooth Engagement — Forces Perpendicular to the Line of Action

36 At this point in the analysis, it would appear that the vibration accompanying premature engagement due to relatively short normal pitch on the driving element is characterized by about the same severity during motoring and regenerating. In actual service, however, we find that, with equal discrepancies, the vibration accompanying regeneration is materially more severe than that occurring during motoring. It is therefore necessary to look further for the reason.

37 With the gears under discussion, the transient peak of normal pressure associated with the impact of premature engagement during regeneration occurs at a point more remote from the pitch point than the corresponding point during motoring. This is due to the long- and short-addendum teeth which are employed. Since the sliding component of the relative motion of mating teeth

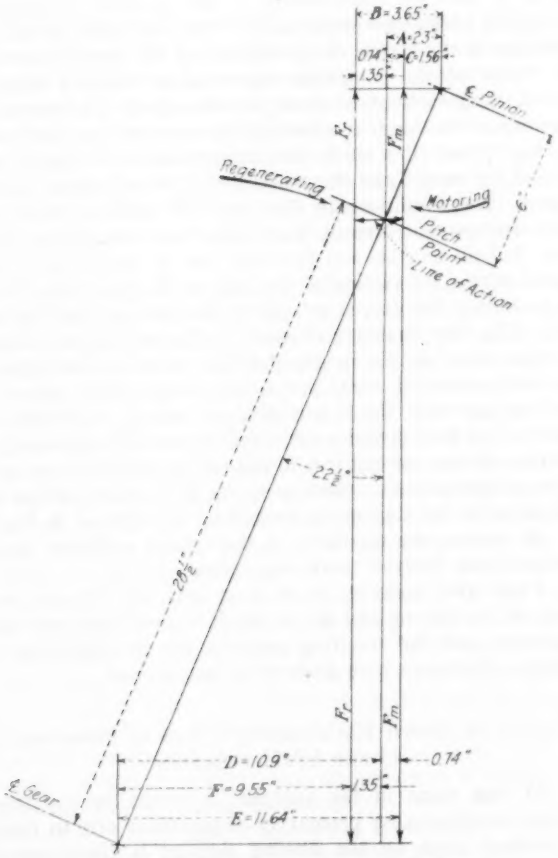


FIG. 10 MOMENT OF TOOTH FRICTIONS

$A = 6 \sin 22\frac{1}{2}^\circ = 6 \times 0.3827 = 2.3$ in.
$B = 2.3 + 1.35 = 3.65$ in.
$C = 2.3 - 0.74 = 1.56$ in.
$\frac{B}{C} = \frac{3.65}{1.56} = 234$

$D = 28.5 \sin 22\frac{1}{2}^\circ = 28.5 \times 0.3827 = 10.9$ in.
$E = 10.9 + 0.74 = 11.64$ in.
$F = 10.9 - 1.35 = 9.55$ in.
$\frac{E}{F} = \frac{11.64}{9.55} = 1.22$

is a direct function of the distance of the contacting point from the pitch point, it follows that the rate of sliding during the premature engagement of regeneration is greater than that occurring during motoring, as shown in Fig. 10, where $\frac{1.35}{0.74} = 1.82$.

38 The product of normal pressure and coefficient of friction gives forces F_m, for motoring, and F_r for regeneration. These forces are directed perpendicularly to the line of action. The coefficient of friction is a function of the condition of the lubricant. This will be referred to later in more detail.

39 Fig. 10 shows clearly that the disturbing turning moment of tooth friction active upon the pinion, and therefore upon the entire rotor, is comparatively small when the pinion drives the gear, being represented by $F_m \times C$, but when the gear drives the pinion, with an identical normal pressure cycle, in the particular gearing under discussion it becomes 2.34 times as great, being represented by $F_r \times B$.

40 It will be noted that the entire load is assumed to be carried by one tooth, even at the ends of the line of action. In new gearing of the class under discussion, this must be expected, as the precision essential to cause an actual division in line with theoretical perfection is too expensive, if at all possible.

41 The following tabulation gives the qualitative tendencies of the couples produced by tooth frictions:

	Pinion	Gear
Motoring:		
Direction of rotation.....	Clockwise	Counterclockwise
Direction of tooth-friction couple	Clockwise	Clockwise
Regenerating:		
Direction of rotation.....	Counterclockwise	Clockwise
Direction of tooth-friction couple	Clockwise	Clockwise

42 With Brown and Sharpe standard tooth forms, the tooth-friction couples discriminate a little less against the regenerating condition, in so far as their maximum value is concerned, though the ratio B/C is less favorable. Here again we face a compromise between the strength of the pinion teeth and the forces inflicted upon those teeth. The final selection appears to be beyond the possibility of pure analytic determination, and can be based only upon broad experience assisted by tests with the torsiometer or other similar instruments. These variable disturbing frictional couples tend to produce oscillations of the entire rotating and associated structures. The study of these oscillations is interesting, though very involved, and would be out of place in this paper. It is sufficient to call attention to the apparent anomaly of friction as a positive force during a part of the cycle. For example, we see that during motoring approach, the turning moment of both normal pressure and tooth friction is clockwise in its action on the

pinion, Fig. 10. This simply means that, through the action of friction, some of the kinetic energy of the armature is transferred to the gear, the armature being negatively accelerated and the gear being positively accelerated.

43 With normal lubrication and under the most severe conditions observed, the oscillations outlined never became sufficiently violent to separate the teeth of the gear and pinion. This was proved by the torsiometer records which reveal an absence of the high frequencies which would be involved.

44 With the teeth of the gear and pinion dry and cutting, however, there is a possibility of extremely severe vibrations being produced by the friction couples. Under this condition, there is also the possibility of the occurrence of higher frequencies, due to the alternate seizing and slipping of the contacting surfaces; that is, a chattering slip. This has received confirmation in service as far as the severity of the vibration is concerned, as dry gears have been found where destructive vibration was promptly quieted by feeding heavy lubricant directly into the mesh.

45 From this analysis, it is obvious that when a large amount of regeneration is required by the service, and particularly when full-load torque is developed while regenerating, the engineer must adopt a shorter normal pitch on the pinion than he would employ if heavy regeneration were not required.

46 The exact best compromise is evidently quite involved, and is again beyond the realm of pure analytic determination. The normal pitches which now appear to give the best all-around results are given later.

Tip Relief

47 A discussion of tip relief in some little detail seems to be unavoidable since the prime reason for the need of tip relief is found in the discrepancies of normal pitch, and also because in setting up a normal-pitch indicator, the radial extent of tip relief must be considered to insure checking the normal pitch at the extreme outer end of the true involute part of the tooth contour.

48 Reference has been made to the tendency existing concurrently with premature engagement to produce permanent deformation of the involute contour. A similar tendency is present with delayed engagement, the tip of the driving tooth and the flank of the driven tooth being subjected to high local pressure over the portion CA of the line of action, Fig. 9.

49 When the conditions surrounding the installation are sufficiently circumscribed to render some degree of tooth wear inevitable, as is the case with heavy-traction electric locomotives, it is evident that the departures from the involute which have been mentioned as possibilities will become actual to an extent largely dependent upon the discrepancies in normal pitch. This wear is

likely to become noticeable first on the flank of the gear tooth at *A*, Fig. 9, for the following reasons:

a With the long- and short-addendum teeth employed, there is, as previously explained, more sliding of the teeth at the recess end of the line of motoring action than at the approach end, and therefore the work of friction is greater at this point. The sliding action consists of the sliding of a relatively long portion of the pinion tooth over a small portion of the gear tooth, thus concentrating the wear on the gear.

b The control of the permanent distortions of the gear during the heat treatment employed on the gearing under discussion, dictates drawing the gear back to a slightly softer condition than is necessary and feasible for the pinion.

c Due to the angles at which the teeth approach each other during motoring and regenerating, the tooth surfaces depart further from tangency when regenerating, and this amplifies the gouging tendency. This is more clearly understood when it is realized that without tip relief the area which is built up by deflection to support the load must be developed on a portion of the tooth contour which lies all on one side of the nominal line of action. Thus, with any combination which permits the slightest premature engagement, the absence of tip relief invites gouging. During sustained regeneration, there is also more tendency gradually to thin the film of lubricant at the critical area of initial contact, due to the sliding characteristic noted.

50 The engineer must therefore consider whether the best overall results will be achieved by adopting the closest available approximation to the involute, trusting that the gears will work out their own correction during service operation, or whether he should approximate those corrections in the manufacture of the gears. We shall first discuss the methods for working out the details of tip relief, and follow with some of the arguments concerning the propriety of its adoption.

51 The maximum working pitch-line velocity of the locomotive gearing used for illustration is about 38 ft. per sec. A normal pitch of the driven gear 0.007 in. shorter than that of the driving gear was found to be too far from equality and caused undesirable vibration, particularly during regeneration. Neglecting elastic relief, we find that at the maximum velocity the time required to travel 0.007 in. is about one sixty-five thousandth of a second. This is not an accurate evaluation of the time involved in completing the adjustment cycle, but it emphasizes the fact that only a very small time element is allowed for the adjustment of the relative angular positions of the driving and driven masses.

52 When the general average discrepancy between the normal pitches of the driving and driven gears *D*, Fig. 5, is known, a simple calculation will determine the length of the interference *E*, Fig. 5.

53 Neglecting the deflections due to the engagement at *A*, if the tip of the driven tooth were formed to match the involute contour of the driving tooth from *G* to *F*, the initial engagement would be smooth, followed immediately by accelerations caused by rolling over the hump at *F* on the driven tooth. This would be less severe than if no tip relief were adopted. The cycle immediately following initial engagement may be further eased off, however, by adopting a form of relief curve that is tangent to the driven-tooth contour at a driven-gear radius less than that of the point *F*, and tangent to the driving-tooth contour at a driving-gear radius less than that of the point *F*. By a similar method, it is equally feasible to work out the tip relief to reduce the severity of delayed tooth engagement.

54 Without going further into the details of the relief curves, it may be stated that with full knowledge of the range of normal-pitch variations as produced by available manufacturing facilities, it is thoroughly feasible to develop tip relief which will enable the mating teeth to make comparatively smooth initial engagement, whether premature or delayed, and which will provide a cam action to lengthen out the time element in which angular adjustments must be made for discrepancies of normal pitch.

55 Knowing the method for determining the proper tip relief, and with a fair conception of its quantitative value, it is logical to analyze any possible objectionable features.

56 Referring to Fig. 4, the line *BG* is the intercept between the two tip circles, on the common tangent to the base circles of the mating gears. It is current practice to divide this distance *BG* by *N*, the normal pitch, and to consider the quotient as representing the number of teeth in contact. Any tip relief will shorten the line of action and increase the percentage of single tooth action (in the case of a one-two tooth engagement). If carried too far, the ratio may become even less than unity, which means intermittent action. For this reason, tip relief has been strongly opposed by one school of engineering thought.

57 Before making the final decision to adopt or abandon tip relief, there are two service conditions demanding attention, which have been intentionally omitted in the previous discussion in order to delay complicating the issue:

a Is the normal life of the pinion equal to the normal life of the gear, or will the gear be called upon to wear out a succession of pinions?

b Will any given pinion run out its life with a single gear, or will the exigencies of the service force it to mate with several gears before it is scrapped?

58 An ancient bone of contention among gear engineers is also involved, which would probably not be mentioned here if the conditions did not permit a rather definite answer, i.e., the hunting-tooth ratio versus the exact multiple or other ratio.

59 Experience with heavy traction electric locomotives shows that the gear will run several times as many miles as the pinions. In the early years of the development of this art, pinions failed from wear and also failed from fatigue breakage, with scattering failures from foreign material in the mesh and other causes.

60 The advance in the art of heat treatment has greatly increased the life of pinions, but fatigue failures are still too frequent, while the retirement of pinions from wear alone has been greatly reduced.

61 This explains the relative thickness of the gear and pinion teeth in the illustrations, the pinion teeth having been increased at the expense of the gear teeth, to give a better balance of performance, as tooth failure from fatigue has been almost unknown with locomotive gears heat-treated by the best modern methods. Further reductions of fatigue tendency are now in sight, but the details are not germane to this discussion.

62 Experience also shows that in a heavy-trunk-line electrification, any pinion must be expected to travel widely among the locomotives of its class, due to the exigencies of the service. The engineer must therefore realize that he is dealing with a system of gears in a broad installation, and that his problem is much more difficult than that involved with a pair of gears that are mated for life.

63 To launch such a system of gearing under the most favorable circumstances, when, as has usually been true to date, the overwhelming proportion of the service is motoring, and when regenerating, though important, is effective for only a small percentage of the total operating time and with reduced torque, the following standard practices must be established:

a The normal pitch of the gear must lie within limits close to the normal pitch of the pinion. It may be slightly greater, and may be less by a considerably larger tolerance

b When regeneration is more important and full-load regenerative torque is developed, it becomes necessary to shorten the normal pitch of the pinion.

64 The functions determining gear performance are so numerous and so involved that a sure determination of the extreme limits for normal pitch which will draw the line between success and failure, may never be sharply defined. But experience is accumulating rapidly in this recently appreciated phase of the gear art which is establishing certain limits as being safe. The doubtful question is how far the limits may be spread in an attempt to relieve the user of the high cost of precision work.

c The tips of the teeth should be relieved to reduce the vibration and noise attendant upon working out relief as a by-product of service operation, and to preserve the involute contour at the flanks of the teeth.

65 The limits to be adopted are necessarily affected by the tendency of the gear and pinion to wear toward or away from equal normal pitch. It is interesting in this connection to note that during the early stages of service in the installation used for illustration, there has been a marked tendency to wear toward equal normal pitch. Therefore the modifications of normal pitch recommended are for the purpose of easing the performance during the breaking-in period.

66 The gear ratio under discussion is 21 to 100, which is a hunting-tooth ratio. The opponents of the hunting-tooth ratio argue that the teeth have no opportunity to settle down, since the No. 1 tooth of the pinion may make a differential approach toward accommodating itself to the No. 1 tooth of the gear, but before these two teeth again come into mating relation, the pinion tooth has attempted to start accommodating itself to the other 99 teeth of the gear. In contrast with this, if the ratio were an exact multiple, as, for example, 20 to 100, each pinion tooth would have to adapt itself to only five gear teeth.

67 A very lengthy argument can be worked up on this question. It is sufficient to state that the arguments for an exact-multiple ratio lose much of their weight when it is recalled that each pinion will probably have to work with several gears during its life. Also, there is small chance that a pinion even temporarily removed from service and then returned to work with the same gear, will be reassembled with the same teeth in register.

Results in Service

68 The first gears and pinions in the installation under discussion were finish-hobbed and then given a drastic heat treatment. During this treatment, the pinions became permanently barreled; that is, the diameter at the mid-width of face was greater than that at the ends.

69 Four major departures from machined condition appeared in the gears after the heat treatment:

a Varying overall shrinkage. (This also occurred in the pinions, but the diameter was small enough to keep this feature within limits that avoided trouble)

b Departure from circle toward an oval and toward more complicated shapes

c Departure from cylinder toward an approximation of a cone

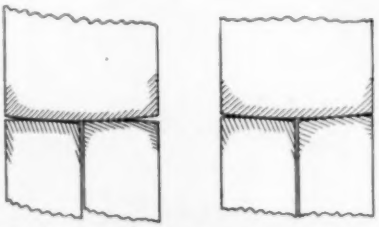
d Departure from plane.

70 A locomotive was assembled in line with previously acceptable practice, with run-of-production gears, and was put into service after a brief test. The effects of certain distortions are shown diagrammatically in Fig. 11. The result was unsatisfactory.

71 A second locomotive was assembled with the gears and pinions matched as regards overall shrinkage, and assembled to

compensate as far as possible for distortions. Fig. 12, again, shows diagrammatically the improvement effected. The preliminary performance showed a material improvement, and this locomotive was also put into service and the gear performance was quite satisfactory.

72 At this stage of the development there was grave doubt as to what constituted the real "measuring stick" for the predetermination of performance. There was, therefore, insufficient assurance of being able to duplicate the action of the matched gears. Arrangements were therefore rushed to grind the contour of the teeth of the remaining gears and pinions. The first ground pinions were produced with facilities that proved inadequate, and their performance was, if anything, worse than that of the unground and unmatched pinions. The first ground gears were produced on a reconstructed gear-cutting machine. All of this grinding was done with a formed wheel, and in the early stages the wheel-trimming devices were far from ideal.



FIGS. 11 AND 12 GEAR RIMS AND PINION ASSEMBLED WITH DISTORTIONS UNMATCHED (LEFT) AND MATCHED (RIGHT)

73 After considerable study, it was decided to base the selection of gears and pinions upon the uniformity of chordal pitch. To be more explicit, a gear was to be tentatively considered good if its chordal pitch was fairly uniform. This chordal pitch might vary slightly from that of the pinion with which the gear was to mate, without condemning the gear or the pinion. The reason for this basis of selection was that the unsatisfactory performances noted were always characterized by some cycle of vibration and noise, which was in phase with the revolutions of the gear. This cycle had decided individuality in many of the gears, and there seemed to be a reasonable expectation that a gear of good uniformity would reduce this cycle within acceptable limits.

CHORDAL-PITCH INDICATOR

74 After gears sufficient for equipping two or three locomotives were ground, the chordal-pitch indicator shown in Fig. 13 and 14 was developed. This indicator has hardened registering points which align it relatively to the side and to the ground bore of the

rim. A hardened pin contacts with a tooth at the pitch line, and a dial indicator with the usual bell-crank attachment contacts with the adjacent tooth at the pitch line. For purposes of record, in inspecting these gears the start was always made on a witnessed tooth. With the instrument in clockwise recording position on the first tooth, the dial was set at zero. With the contact pin in the same space, the indicator was swung to the other hand and the counterclockwise calibrating setting at zero adjusted on the second dial indicator. The indicator was then entered on the next tooth and the readings were recorded as No. 1. All teeth were measured and recorded successively, the hundredth readings being

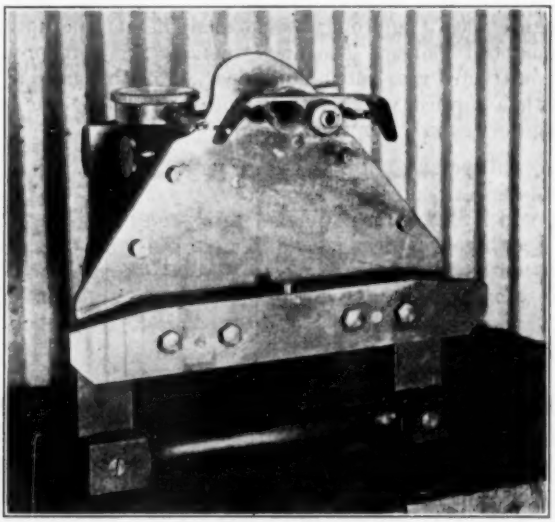


FIG. 13 CHORDAL-PITCH INDICATOR

taken on the tooth of original dial adjustments, which must therefore read zero and check that the calibration of the indicators had not been altered. The readings were then plotted and the datum line corrected to read the average.

75 It will be seen that the plotted values were the variations from tthe average chordal pitch and did not record the actual pitch values.

76 All the gears that had been ground up to this time were plotted, and it was found that the best that could be expected from the facilities then available was a variation from average chordal pitch of ± 0.004 in. This value was therefore adopted as the limit for the locomotives involved. Every tooth that exceeded this limit was retrimmed on the machine and brought within the limit. As the machine aged, it was necessary to regrind some gears.

77 The unsatisfactory unmatched and unground gears and pinions on the first locomotive were replaced by gears selected by means of the chordal-pitch indicator, and within the limits noted. Their performance was perhaps a little inferior to that of the unground but matched gears of the second locomotive. This, of course, was very disappointing.

78 In the meantime, another gear-grinding machine was being brought through by the Newark Gear Cutting Machine Company. As soon as this machine was installed, it was tried out; and it was found practicable to cut the limit of departure from average

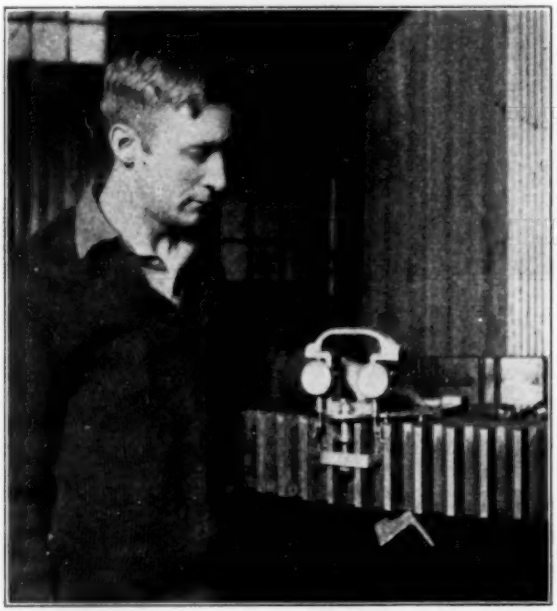


FIG. 14 CHORDAL-PITCH INDICATOR

chordal pitch to ± 0.002 in. A trimming device was also made by the same company, which gave much better contours.

79 The third locomotive was equipped with gears and pinions within this limit. The preliminary trials of this locomotive showed that the gearing was still unsatisfactory.

80 These events led to the analysis outlined in this paper, which shows the importance of a proper relation of the normal pitch of the gear and the pinion.

NORMAL-PITCH INDICATOR

81 The normal-pitch indicator shown in Figs. 15 and 16 was then developed. This outfit included segments of the base circle

bolted to the gear in proper concentric relation, and a tangent frame. This frame determined the location, upon the common normal of adjacent teeth, of a dial-indicator finger at the outer end of the involute contour of one tooth, and of a locating plug on the adjacent tooth. The dial was set to a calibrating plug gage to read

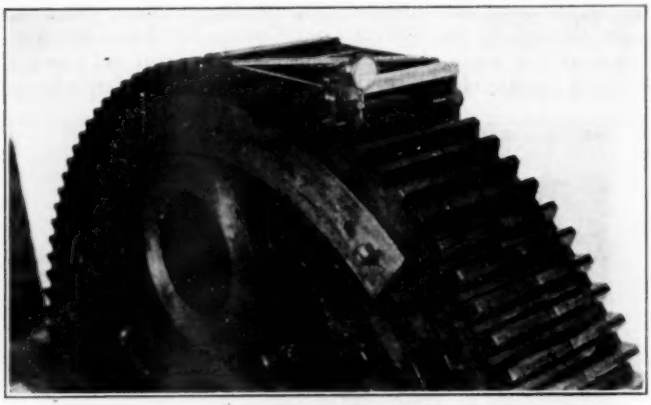


FIG. 15 NORMAL-PITCH INDICATOR

zero at the exact calculated normal pitch. This indicator therefore read the actual normal pitch instead of a relative value as in the chordal-pitch indicator. This indicator was found to be very clumsy to handle, and it had the serious limitation of all indicators developed to this time; namely, it could be applied only at lateral locations about an inch away from the edge of the gear.

FIG. 16 NORMAL-PITCH INDICATOR

82 It was useful, however, in the practical development of the fact that variations of plus or minus a sixteenth of an inch in the radius of the base circle produced errors so small as to be within the accuracy of reading of the indicator. As this variation more than covered the range of permanent shrinkage of the gears during heat treatment, it was at once evident that a very convenient

instrument could be made which would read at any point across the face of the gear. This instrument could register on either the tips of the teeth or the bottoms of the tooth spaces, or the registering parts could bottom on the flanks of two adjacent teeth. The

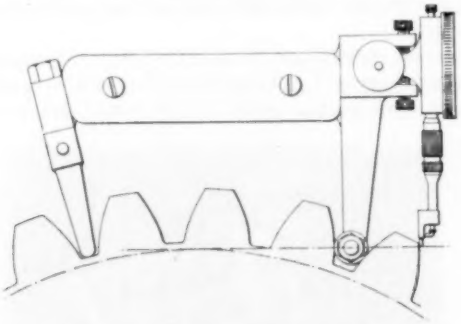


FIG. 17 NORMAL-PITCH INDICATOR

combination shown in Figs. 17, 18, and 19 was adopted. A hardened and ground cylindrical member bottoms on the flanks at a depth which brings its contact on that common normal to two tooth contours which passes through the outer end of the involute part

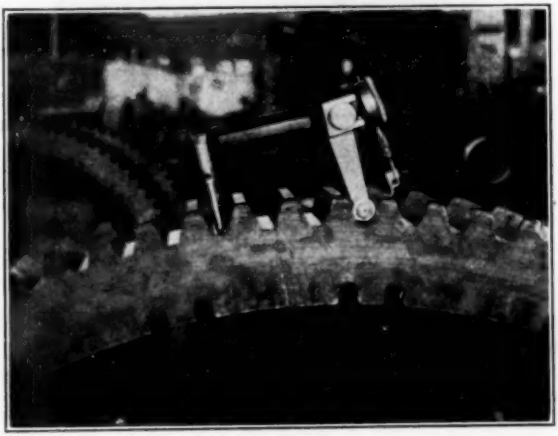


FIG. 18 NORMAL-PITCH INDICATOR

of the contour of one of the two teeth, at which point an indicator finger is arranged to make contact. The alignment of this common normal is determined, within the required accuracy, by the locating leg remote from the indicator and registering in the bottom of a

tooth space. The length of the contact cylinder is sufficient to insure a firm, square setting of the indicator.

83 This indicator is made adaptable to all $1\frac{1}{4}$ diametral-pitch gears between 50 teeth and a rack, by means of a range of contact cylinders of suitable diameters, and a range of locating legs of suitable lengths.

84 A second normal-pitch indicator was developed covering the remaining range of $1\frac{1}{4}$ diametral-pitch gears and pinions. This particular type of normal-pitch indicator may become unreliable

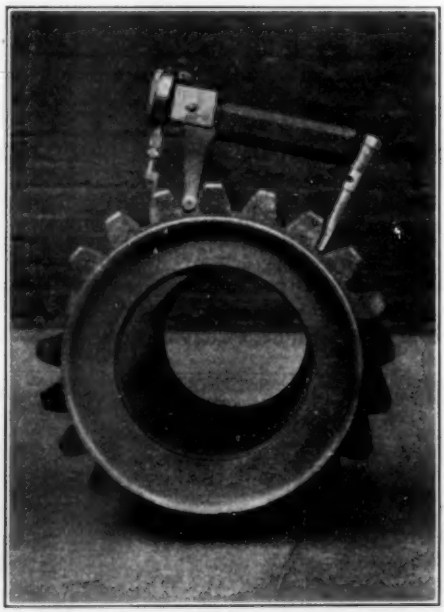


FIG. 19 NORMAL-PITCH INDICATOR

for pinions with less than, say, 17 or 18 teeth, as the line of indication may depart too far from the line of action.

85 The check-up on the range of normal pitch of the gears showed that they ran quite uniform, the variation being about 0.004 in.

86 The pinions also were quite uniform, the variation being about 0.002 in., but the actual value of their normal pitch was about 0.007 in. longer than that of the gears, and this was the principal reason, from the gear standpoint, for the unsatisfactory operation. While the motoring operation was better than it would have been had the normal pitch of the pinions been shorter than that of the gears by the same amount, it was too far from equality.

87 A set of pinions was produced with a normal pitch 0.002 in. shorter than the gears, and while this discrepancy was still on the less desirable side of equality for the motoring condition, it proved to be about the best compromise for the overall service of motoring and regenerating, and corrected the difficulty that had been experienced.

88 A considerable range of installations using gears and pinions as they clean up in grinding out heat-treatment distortions, is under way and is developing in some measure the limits of dis-

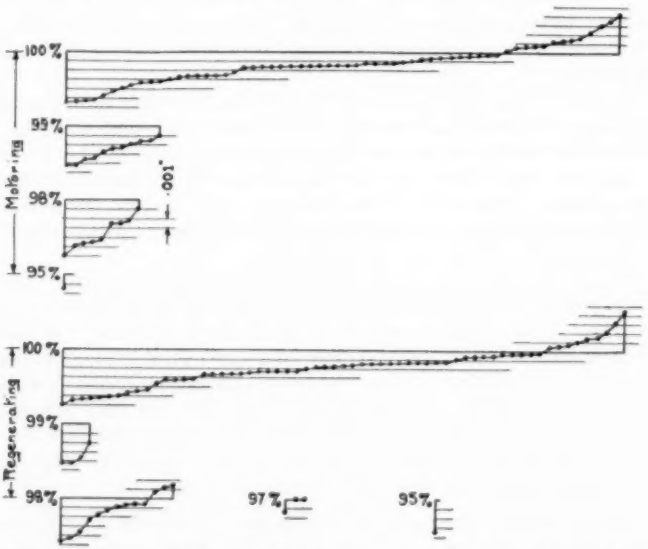


FIG. 20 INFLUENCE OF NORMAL PITCH ON GEAR PERFORMANCE ON TEST TRACK

Percentage indicated is rating of gearing on test track. Each point represents a gear and pinion test. Distance below base indicates normal pitch of pinion is less than that of gear.

crepancy of normal pitch within which acceptable operation can be achieved with the particular gearing and service under discussion. The results of this class to date are outlined below.

ANALYSIS OF INFLUENCE OF NORMAL PITCH ON GEAR PERFORMANCE ON TEST TRACK

89 All gears after installation in the completed locomotive are given a performance rating on the test track. This rating is based upon noise as measured by the ear of a trained observer, and amplitude of vibration as judged by the hand of the same observer. The parts checked for vibration amplitude are brushes, brush-

holders, and leads. The gears giving the best performance are rated 100 per cent. The lowest passing grade is 95 per cent.

90 This method of rating is open to some criticism as it is not possible for the observer to calibrate himself with complete accuracy. Continual care is required on the part of the observer to distinguish between noise and vibration, as it seems to be pretty thoroughly established that they do not bear a constant relation. Vibration is therefore the major consideration in rating the performance.

91 Instruments are being devoloped, with which it is expected to be possible to secure quite an accurate record of the amplitude of vibration in a manner thoroughly practical for commercial inspection.

92 In each of the graphs of Fig. 20, the base line indicates an equality of the normal pitches of the gear and pinion. Values below the base line indicate that the normal pitch of the pinion is less than that of the gear by the number of thousandths of an inch plotted. Values above the base line indicate that the normal pitch of the pinion is greater than that of the gear by the number of thousandths of an inch plotted.

93 Reference to the graphs shows that satisfactory performance has been secured over the entire range from minus six thousandths to plus four thousandths as the value of the expression, "normal pitch of pinion minus normal pitch of gear." This is roughly true under the conditions of both motoring and regenerating. Practically all performance of a grade less than 100 per cent occurred with the normal pitch of the pinion less than that of the gear.

94 Taking the graphs at face value, any one of the following conclusions might be drawn:

a The theory advanced in this paper is entirely wrong, and normal-pitch relation has little or no influence on performance within the limits of accuracy usually achieved in cut gearing

b The touch-and-ear method of rating and performance of gearing is too approximate to permit close analysis of the results

c Other variables must be considered jointly with normal pitch to foretell reliably the performance which it is logical to expect from a gearing installation.

The first conclusion would be superficial in view of the common sense of the theory. The second and third are sound, and account for the conditions shown in the graphs. Pending the availability of the instruments to which reference has been made, it is illogical to pursue the analysis of the graphs further.

95 The outstanding fact of this study is that only with a high degree of accuracy in the entire mounting can the variations of normal pitch be permitted to go as high as indicated in the graphs. A considerable number of mountings were encountered in the early stages of the work where departures from accuracy of normal

pitch gave unsatisfactory performance, and these cases were corrected by following the tendency of the theory in the proportioning of the gear and pinion. During the latter stages of the work, the accuracy was held well within the limits indicated, and nearly every gear and pinion was inside the satisfactory performance range on its first trial.

Errors of Mounting

96 No detailed reference has been made of the errors of mounting that were found to exist, as simplicity was served by confining attention to the gear teeth. Concurrently with the investigations of pitch, the entire mounting was overhauled and checked for accuracy, and the existing errors reduced as far as practicable. The overall investigation proved conclusively that acceptable operation could be secured only when the discrepancies of both the mounting and the gears were held within proper limits in their salient characteristics.

Conclusions for Locomotive Motoring and Regenerating Service

97 The conclusions for locomotive motoring and regenerating service are:

a The normal pitch of the driving and driven gears must lie within limits determined on the basis which has been set forth.

b The tips of the teeth should be relieved on the basis outlined. This is more vital on the pinion than on the gear.

c Acceptable gear operation can be secured in locomotive installations where back-driving is essential.

d The degree of quietness of gear operation can be fairly well predetermined by checking the known salient characteristics of the gears and mounting.

e Selection can be made between gears that require grinding and those that may successfully be put into service as they come from heat treatment.

f An adequate supply of proper lubricant must be present on the teeth during the entire working cycle of the locomotive.

g Data must be accumulated for a considerable period to determine the widest limits that may be adopted for the salient variables in order to make costs as reasonable as possible.

DISCUSSION

L. F. Burnham.[1] The writer wishes to point out certain limitations in the application of the theory and certain amplifications and modifications thereof. He would emphasize that the principal

[1] Mechanical Engineer, R. D. Nuttall Co., Pittsburgh, Pa. Mem. A.S.M.E.

application of this theory should be to new gearing. It is believed possible to predict with some degree of accuracy that certain gear applications will wear out due to their tooth proportions, speed, loading, mounting, and the moving masses connected therewith, while other gear applications will tend to wear in. By wearing in, it is understood that all normal pitches in the gear system tend to be equalized and to approach the theoretically correct normal pitch. The principal reason for wearing in is that high spots tend to be worn off and contour depressions to be passed over when there is more than one tooth in contact. On the usual gear set, wear will soon bring more than one gear tooth in contact, if this is not at first the case. A reason for wearing out, as the opposing tendency, is the combination of high sliding velocity with heavy pressures in one portion of the tooth as compared with another portion, or in other words, unequal frictional work in different parts of the contour.

The principal application of the new theory in new gearing must be to the ground gear, and in particular to large gears. Use of the theory rests on the degree of accuracy obtainable with machine tools. The theory must be precluded, therefore, unless it is possible to control tolerances within rather close limits. This is a difficult problem with the best gear-cutting tools and not considered feasible at present. In the case of grinding, constant attention must be paid to the grinder, the abrasive wheel, and the trimmer to keep the tolerances in the direction and of the amount predetermined. It is believed that automotive gearing when ground is susceptible of much greater accuracy than gears of the size under discussion, due to the relative sizes and to the machines at present available. This would obviate the necessity for modifications of normal pitch as called for in this theory. Any class of gearing to which it is desired to apply normal-pitch modifications should be studied from such points of view as to determine if modifications are desirable.

It is believed desirable to define normal pitch as used in this paper. Algebraically it is the circular pitch multiplied by the cosine of the pressure angle, i.e., an exact measuring stick. Geometrically it is the theoretically correct distance between two adjacent gear teeth along a common normal to their involute contours. These normals need not be along the line of action of a gear set, but they will coincide therewith when the gear teeth are in contact on the normal under consideration.

Several conditions may be present in the tooth contours when normal pitch is not theoretically correct:

1 The tooth may be displaced circumferentially; that is, the contours of the teeth are involutes with respect to the base circle, but the teeth are unevenly spaced. Such a displacement would be caused by indexing errors.

2 The contour of the tooth may be some other curve than an involute. For instance, it may be heavy at the tip and narrow at the root, when long normal pitches would be measured. With the conditions reversed, that is, relief at the tip and heavy roots, short normal pitches would be measured. The contour might be irregular, when either short or long normal pitches would be measured. The first two conditions might be caused by use of form cutters, set to incorrect depths. The last condition might be caused by irregularities in cutters, etc.

Numerous conditions could and do conspire to bring about the conditions described, whether the teeth are formed or generated. All gears contain these errors in some degree.

The effects mentioned in this paper are not the only ones encountered. For instance, tip relief, above mentioned, will cause acceleration in the gear set during contact over the remaining contour.

With respect to the "interferences with smooth engagements," as mentioned in Par. 19, the writer believes that deflection is of a secondary order of importance. High spots are fractional parts of a thousandth of an inch. Surface depressions with usual gear loads and radii of curvature, minor and cantilever deflections are also small. For instance, calculating a $1\frac{3}{4}$-pitch tooth as used in locomotive service, with a 1500-hp. motor at 362 r.p.m., the cantilever deflection amounts to 0.00028 in. Assuming that all three reasons for surface depression amount to 0.0005 in., machining errors are of primary importance. These errors will be at least ± 0.002 in., or eight times the variation due to deflections.

In Fig. 6, the line of action proceeds along *BCA*. At first this would appear incorrect as it is not along the normal to the tooth contours. Where such action occurs, the conception may be clarified if it is considered that as contact cannot occur on the top land of a tooth, there is a dwell of contact on point *B*, considered as a point in gear until *B* has rotated with the gear to position *C* along the addendum circle. It is also well to point out here that this dwell must be a position of pure geometric sliding on the pinion contour, with high frictional work and rapid wear on the gear tip. This condition is reversed in Fig. 9, the wear here coming on the pinion tip, but the relative length of contours to the pitch point will vary the rate of wear at different points along the length.

It may be noticed that the best compromise for motoring and regenerating is indicated in Par. 87, where the pinion normal pitch is given as 0.002 in. shorter than the gear normal pitch.

Referring to Par. 24, this would appear to be opposed to the theory of a long normal pitch on the driver. It is perhaps superfluous to point out that the shorter normal pitch of the pinion as compared with the gear is confirmatory of the theory, since regeneration proved to have been the controlling direction of drive.

The writer's company is proceeding further with the design and manufacture of suitable tools for measuring normal pitch. The tool roller shown in Fig. 17 sinks to different depths in the tooth space, depending on the backlash. While this is not serious, as the desired backlash can be allowed for in the roller diameter, it is felt desirable to eliminate any consideration thereof. It also happens, depending on the gear size, that the normal to the involute contour along which it is desired to measure will require a flattened roller to prevent striking the root of the tooth. The latest tool therefore has two legs which rest in the forward space and one leg in the rear space on the gear root circle. These three legs provide stability. A sharp edge is located in the correct position of contact with the involute on the normal and is fastened to the two forward legs.

This edge can be rotated angularly about the gear center by moving the instrument backward in the spaces until it makes contact with the involute contour of the tooth. This eliminates consideration of backlash, never of primary importance. It also prevents the use of flattened rollers. It can be seen that the reading of the instrument, properly set, does not depend on the skill of the operator but is inherent in the design of the tool.

B. F. WATERMAN.[1] The writer is impressed with the emphasis the author lays on the variables that affect quiet running. Not only does he discuss the shape of the teeth and their variables, but he lays great emphasis on the gear mounting. Usually, when a pair of gears are noisy, the first thought is that they have been poorly cut or ground (if the teeth are ground), but rarely does the mounting receive consideration. The author's analysis of "tooth engagement" is instructive and the conclusions drawn, in the writer's opinion, are correct.

It has been the practice of the writer's company for some time to state on gear drawings, as far as is possible, all facts concerning the gears. For instance, besides the number of teeth, pitch, pressure angle, and the method to use in cutting, there are given the r.p.m., the gear with which it runs, whether it is driven or driver or both, its surface speed, and the load, if this can be calculated with any degree of certainty. The reason for this is that if the gear-cutting department knows what the gears are to do, it can take the necessary precautions to produce the kind or quality of gear desired.

It also has been our practice on gears for severe service to see that the proper contact is obtained as outlined by the author. That is, a more prominent or higher bearing on the teeth of the driving member and a lower one on the driven gear are provided

[1] Designer, Brown & Sharpe Mfg. Co., Providence, R. I. Mem. A.S.M.E.

to prevent abrupt contact on the approach. When both members are drivers, our problem, or rather the result we would desire, would be not unlike that which the author finds the best and which he mentions in Par. 87.

We have always felt the desirability of tip relief, and incorporate it in most of the cutters and hobs that we make.

H. J. EBERHARDT.[1] The subject of the influence of a longer pitch on the driving member of a pair of gears is a very old one. In earlier days it was common practice to make pinions having a ratio of four or five to one about 1/32 in. larger in diameter for a pitch diameter of about 4 in. Good results were obtained thereby. As formed cutters are now better made, it is found that worse results may be obtained by making the pinions larger in diameter. In reviewing the history of troublesome gears, we find errors in the circular pitch division, as well as in the form and non-uniformity of the form on each side of the gear teeth.

The remark by the author that these gears are much more crude than automobile gears is open to comment, as it should be realized that the larger of this pair is about five feet in diameter, say, 57 in., and the error is less than ± 0.002 in. in the spacing and normal pitch. On a gear one foot in diameter it would be about 0.0003 in. The gears described in the paper are being made under conditions that are somewhat comparable to laboratory conditions.

If we were to set a limit of ± 0.001 in. on a 5-ft. gear, in division and normal pitch, we would have a difficult proposition. It would make the gear cost more than twice as much in the gear-tooth-forming part of the operation. The difference between 0.001 in. and 0.002 in. is 100 per cent.

There are several important variables that come into the problem, which control the tooth form and the spacing on this very accurate gear work. The error of ± 0.002 in. is the result of an accumulation. There is the possibility of error in the form of the grinding wheel. This is a complex problem, running into the wear and breakage of diamonds used in forming the wheels. Then the gear-grinding machine itself must be kept nicely adjusted in the grinding-wheel spindle bearings, the work spindle bearings, the mechanism controlling the index wheel, and the flexible part of the indexing train whereby different numbers of teeth are indexed. If such a machine were made to work like gear-manufacturing machines in aeronautical and automotive service, several of the error possibilities would be eliminated. The gears being discussed were made on a commercial scale under average machine-shop conditions, and the men making them really were doing

[1] Secretary and Engineer, Newark Gear Cutting Machine Co., Newark, N. J. Mem. A.S.M.E.

laboratory work unconsciously. One thing should be emphasized. In work like this, regardless of the refinements of the tools and gages, the human equation will always be of prime importance. Good gears are the result of continuous care and vigilance.

R. E. FLANDERS.[1] The conception of the normal pitch is a useful one. This is the only characteristic of an involute gear which is born with it and remains the same to the end of its life. Put two involute gears together, and the diametral pitch, the circular pitch, the pressure angle, and the pitch diameter all change, depending on the center distance at which the gears are run. Nothing remains the same except the normal pitch. That is the element that tells what the gear is, and from this standpoint it is a most useful designation.

Can the author tell us if there is any possible advantage in making a deliberate slight difference in normal pitch between the driver and driven gear with the idea of correcting thereby for the deflection of the teeth under the normal load to which they are to be subjected? It would seem that such a difference of the proper amount, while giving under no load a delayed contact, under normal load would give more nearly a normal contact than would otherwise be possible.

S. TIMOSHENKO.[2] It may be of some interest to know about the deflection of gear teeth. The author divides this into three types: first, flattening down of high spots; second, depressions on the surface and contour; and, third, deflections caused by the bending of a tooth. The first type of deflection is difficult to figure, because it depends on the conditions of finish of the surface. The deflection due to depressions on the surface of contact can be figured by Nomhart's theory. If this be applied to the case discussed in the paper, it will be but a fraction of 0.001 in.

In regard to the deflection of a cantilever beam, if this deflection be figured on the basis of a simple cantilever, in the case of locomotive gears and with pressure of 2000 lb. per in. of face of gear, deflections in the neighborhood of 0.0005 in. will be obtained. In order to get more reliable data on the deflection of gear teeth, we have made direct tests of deflection of this kind, which showed that the cantilever-beam formula gives a fairly accurate result. The variation in cross-sections and the effect of shear were taken into consideration. The experiment gave only ten per cent greater deflection than the calculated deflection, but it showed that the deflection due to bending of the gear rim is of importance. In the

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[2] Research Engineer, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa. Mem. A.S.M.E.

case of a comparatively thin rim, and a large diameter of the gear, as in locomotive gears, the deflection will be much larger than the deflection of the gear tooth as a beam. The deflection will occur in the gear and not in the pinion.

There always is a very high local pressure at the beginning of engagement, and even in the case of smooth engagement. Hence some rounding of the tooth will be very useful.

THE AUTHOR. Mr. Flanders asked if we can actually produce the modulation of normal pitch. We can and do. At first we modified the normal pitch in order to favor the motoring condition, because we did not realize that the regenerating condition was somewhat more difficult than the motoring condition, for reasons brought out in the paper. Also, in our previous experience we have had less regeneration than motoring, and what we did have was at a lower tractive effort. Later we had prolonged regeneration and discovered that regeneration was the commanding position. Under these conditions we had to shorten the normal pitch of the pinion.

Mr. Burnham referred to normal pitch, defining it as a theoretically correct, definite measurement, and then in his discussion he said "when normal pitch is not correct." He evidently had in mind the same definition of normal pitch that the author had. It is the thing that exists on the gear. It is not the normal pitch of a gear system, but the normal distance between the faces of adjacent teeth as they actually occur on a given gear which is up for investigation.

No. 1974

TORSIONAL-STRESS CONCENTRATIONS IN SHAFTS OF CIRCULAR CROSS-SECTION AND VARIABLE DIAMETER

By Lydik S. Jacobsen,[1] Stanford University, Cal.

Non-Member

The object of this paper is to point out an electrical experimental method for finding the torsional-stress distribution in cylindrical shafts of circular cross-section and of any axial outline. The method has been applied to the problem considered by F. A. Willers, and the paper includes a series of curves that will enable the designer to find, at a glance, the maximum torsional stress in a shaft of two diameters for various diameter and fillet proportions. It should be remarked that the results for a shaft of two diameters may be applied to most shafts of several diameters since the length of any two adjacent portions of the shaft are usually sufficiently long to make the analysis apply. Special shaft outlines, for instance shafts with oil throwers, can be investigated by the method.

Introduction

AFTER the proportionality law had been formulated by Hooke, a number of investigators became interested in problems of elasticity. One of these problems was the torsional rigidity and strength of bars. Making the assumption that no external forces act on the sides of the bar, and that cross-sections of the bar originally plane and perpendicular to the axis remain plane and perpendicular after twisting, Coulomb in 1784 published the first rational theory of torsion. Since then the results obtained by Coulomb have been applied generally by the engineering world. In fact, the extension of the simple torsion analysis became so general that there was a misapplication in many instances.

2 Cauchy was the first to apply the general equations of the theory of elasticity to torsion problems and to show that Coulomb's

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analysis is valid only for the case of a bar of circular cross section. The reason for this limitation of the analysis is that a bar of circular cross-section is the only type of bar in which cross-sections originally plane remain so after twisting. At that time, in 1828, it was also realized that whenever the cross-section of a bar changes abruptly in the axial direction, as in the case of a cylindrical shaft of circular cross-section and of two diameters, torsional-stress concentrations may be expected in the region at which the change of cross-section occurs. It has been generally known for some time that in shaft design sudden changes in the shaft diameter should be avoided by allowing a generous fillet at the place where the shaft diameter changes. Introducing a fillet at such a place will diminish the stress concentration which invariably occurs there. Whether the loading of the shaft be axial, transverse, or torsional, the maximum stresses corresponding to each form of loading will be higher at the place where the diameter change occurs than in the straight parts of the shaft adjacent to this place.

3 A survey of technical literature discloses that little information is to be had about the actual magnitude of such stress concentrations. The mathematical analysis of the torsional-stress problem has been investigated by several writers, but the solutions offered involve extremely laborious graphical solutions by successive approximations.[1] In the dissertation written by Herr Willers, the graphical solution of the problem is developed fully, and a curve is given showing the maximum torsional stress in a shaft of two diameters for various fillet sizes. The amount of painstaking work necessary to determine the four points through which the curve is drawn is very great, so that the method cannot readily be adopted by designers for finding stresses in shafts of somewhat different proportions.

4 Before proceeding any further, it may be well to point out the practical importance of a knowledge of the theoretical stress concentration in a shaft. Assuming that it can be shown that the theoretical maximum stress concentration in a shaft corresponds to a stress of twice the elastic limit of the shaft material, will the shaft fail?

5 Since the theoretical analysis is based on Hooke's proportionality law, it must be admitted that in the cases where the theoretical stresses are higher than the elastic limit, the actual stresses cannot be as high as the theoretical; in other words, the material will yield in deformation and thereby allow a stress distribution different from the theoretical. If the material be sufficiently ductile to permit an elongation adequate to transfer the stress from the highly stressed regions to the less stressed, the shaft will not suffer immediate failure. We say immediate failure be-

[1] F. A. Willers, *Zeitschrift für Mathematik und Physik*, vol. 55, 1907, pp. 225-263.

cause all ductile materials will withstand a certain number of repeated deformations beyond their elastic limits. If, however, the character of the applied twisting moment be variable, the stress and the deformation will vary, and a fatigue failure is very likely to result in the course of time. In a number of instances such failures have occurred several years after the shafts had been put into service.

6 The object of this paper is to point out an electrical experimental method for finding the torsional stress distribution in cylindrical shafts of circular cross-section and of any axial outline. The method has been applied to the problem considered by F. A. Willers, and the paper includes a series of curves that will enable the designer to find, at a glance, the maximum torsional stress in a shaft of two diameters for various diameter and fillet proportions. It should be remarked that the results for a shaft of two diameters may be applied to most shafts of several diameters since the length of any two adjacent portions of the shaft are usually sufficiently long to make the analysis apply. Special shaft outlines, for instance shafts with oil throwers, can be investigated by the method.

Simple Qualitative Method of Discussing the Problem

7 In order to understand the torsional problem and its electrical analogy without going through the mathematical analysis,[1] the following elementary considerations have been made.

8 Coulomb found that in a cylindrical shaft of circular cross-section subjected to pure torsion, the torsional stresses vary directly as the distance from the axis of rotation. Furthermore, that the angle of twist per unit length of the shaft is inversely proportional to the polar moment of inertia of the cross-section of the shaft, i.e., inversely proportional to the fourth power of the radius of the shaft. Making use of this information, we can imagine a shaft divided into a number of concentric cylinders, with the understanding that each cylinder is to carry the same fraction of the total twisting moment.

9 Fig. 1 shows a shaft that has been divided into five concentric cylinders, each of which carries one-fifth of the total twisting moment. It will be noticed that the thicknesses of the cylinder walls are different, in fact, the thicknesses have been chosen in such a way that each cylinder, when it is subjected to one-fifth of the total twisting moment, will have the same angle of twist θ per unit length as the other four cylinders. From this it follows that there will be no relative displacement between the cylinders when a twisting moment is applied; in other words, the five cylinders will rotate through the angle of twist as a single body. Since the cylinders rotate as a single body, fictitious planes equal dis-

[1] Given in Appendix.

tances apart and perpendicular to the axis of rotation will contain all the points that have rotated through a certain angle of twist with respect to an adjacent plane. Fig. 1 shows the fictitious planes as well as the concentric cylinders. For the sake of generality, let us call the cylinders "equimomental tubes" and the planes "equiangular surfaces." In order that the following may be clear it is important that a physical conception of the above division of the shaft have been acquired.

10 Let us consider a shaft of two different diameters connected by a fillet. It is evident that in regions of the shaft sufficiently distant from the fillet, the division of the shaft into equimomental tubes and into equiangular surfaces will be as shown in Fig. 1. In the neighborhood of the fillet the division will be different from the one in Fig. 1; the equimomental tubes will no longer be simple cylinders, and the equiangular surfaces will no longer be planes.

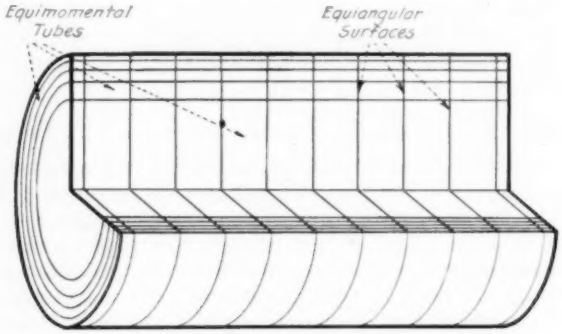

FIG. 1

The equimomental tubes and the equiangular surfaces will, however, be drawn in such a way that they continue to intersect at right angles. The conditions that determine the location of the equimomental tubes and equiangular surfaces are:

1 The condition of torsional equilibrium
2 The condition of continuity.

The two conditions are intimately connected, but in a qualitative analysis the first cannot be taken into account. After the condition of continuity has been satisfied, we get a qualitative picture of the equimomental tubes and equiangular surfaces as shown in Fig. 2.

11 Let us assume that Fig. 2 is a correct two-dimensional representation of the equimomental and equiangular division. We will now consider the torsional equilibrium, i.e., the relation of the tubes and surfaces to the relative torsional stresses. By definition an equiangular surface has the property that it contains

all the points that have been rotated through a certain angle of twist θ with respect to the points in another equiangular surface when a twisting moment has been applied. Calling the angle of twist $\Delta\theta$, it is seen that all points lying on surface *ab* have been rotated through the angle $\Delta\theta$ with respect to the points lying on the surface *cd*. Similarly, by counting the spaces on Fig. 2, it is seen that the points on surface *gh* have been rotated through an angle of $17\Delta\theta$ with respect to the points lying on surface *ab* or through an angle $\Delta\theta$ with respect to the points on surface *ef*.

12 Consider the part of one of the equimomental tubes lying between the equiangular surfaces *ab* and *cd*. In Fig. 2 the part of the tube under consideration has been cross-hatched. A twisting moment, one-fifth of the total twisting moment, is being transmitted by the tube, and the angle of twist $\Delta\theta$ will be established

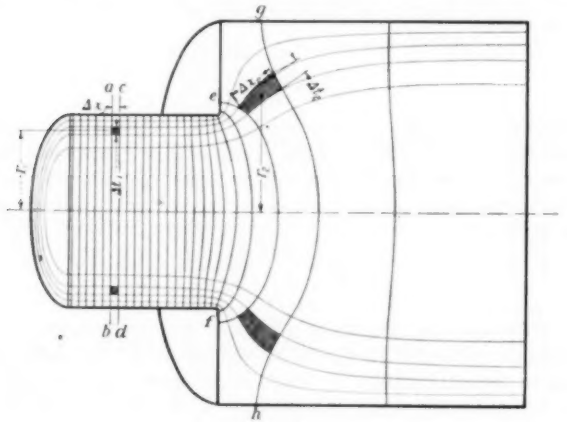

FIG. 2

over the length Δx_1. If the mean radius of the tube be r_1, the approximate polar moment of inertia of the cross-section of the equimomental tube under consideration is

$$I_p = 2\pi r_1^3 \Delta t_1 \quad \ldots\ldots\ldots\ldots \quad [1]$$

Using Coulomb's simple torsion formula to express $\Delta\theta$, we easily obtain

$$\Delta\theta = \frac{\Delta M \Delta x_1}{2\pi r_1^3 \Delta t_1 G} \quad \ldots\ldots\ldots\ldots \quad [2]$$

where G is the modulus of shear of the material. Applying a similar process of reasoning to the part of the same equimomental tube that is contained between the equiangular surfaces *ef* and *gh*, we obtain

$$\Delta\theta = \frac{\Delta M \Delta x_2}{2\pi r_2^3 \Delta t_2 G} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [3]$$

where the subscripts denote the new quantities. Since ΔM and $\Delta\theta$ are the same in Equations [2] and [3],

$$\frac{1}{r_1^3} \cdot \frac{\Delta x_1}{\Delta t_1} = \frac{1}{r_2^3} \cdot \frac{\Delta x_2}{\Delta t_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [4]$$

Now, since the tube in question carries the same twisting moment throughout,

$$2\pi r_1^2 \Delta t_1 T_1 = 2\pi r_2^2 \Delta t_2 T_2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [5]$$

where T_1 is the average torsional stress in the equimomental tube of thickness Δt_1, and T_2 is the average torsional stress in the tube of thickness Δt_2. Rearranging this equation and cancelling, it follows that

$$\frac{T_2}{T_1} = \frac{r_1^2}{r_2^2} \frac{\Delta t_1}{\Delta t_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [6]$$

Combining Equations [4] and [6], the ratio of the torsional stresses at the two points under consideration, expressed by the distances between the equiangular surfaces, the lengths Δx_1 and Δx_2, and the radii r_1 and r_2 (a more convenient form as will be shown later),

$$\frac{T_2}{T_1} = \frac{r_2}{r_1} \frac{\Delta x_1}{\Delta x_2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [7]$$

13 By means of Equation [7] and a correct layout of the equiangular surfaces, the stress at any point of the shaft can be calculated approximately. It must be remembered that the solution is approximate when dealing with finite distances of Δx_1 and Δx_2, if the angle $\Delta\theta$ could be made infinitesimal, Δx_1 and Δx_2 would become infinitesimals, and we should have an exact solution. Fig. 2 shows that the distances of Δx are shorter in the neighborhood of the fillet than at other places, indicating that a stress concentration exists in the fillet region.

14 We are now confronted with the problem of making a correct layout of the equiangular surfaces. To obtain a general solution of the partial differential equation governing the location of the equiangular surfaces is not difficult, but due to the fact that the boundary conditions to be satisfied are in most practical cases rather cumbersome to handle analytically, the particular solution of the problem becomes extremely complicated. The general solution of the problem shows that the relation expressed by Equation [4] in the preceding simple analysis must be fulfilled in the layout of the surfaces. A graphical layout would therefore constitute an approximate solution of the problem. All that is

necessary is to draw the outline of the shaft and to trace the two-dimensional representation of the equimomental tubes and equiangular surfaces, making sure that the relation expressed by Equation [4] is satisfied for every rectangular cell formed by the two systems of lines. In plain language Equation [4] means that the length of a cell divided by the width of the same cell is proportional to the cube of the radial distance the cell is away from the axis of rotation. However simple this may seem, it is a fact that it requires a great deal of patience to trace the two systems of lines if a tolerable degree of accuracy is desired. It was mentioned in the introduction that F. A. Willers has developed a method of tracing the lines, a method of successive approximations which finally converges toward the correct solution, but even with this great help the amount of work is considerable.

An Electrical Analogy of Torsion

15 An experimental method of obtaining the correct layout of the equiangular lines will now be discussed.

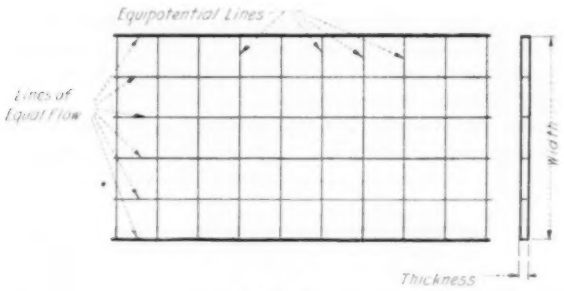

Fig. 3

16 Consider the flow of electricity in a long strip of metal. If the strip be of constant width and thickness and of constant conductivity throughout its length, the electrical potential will fall at a uniform rate in the direction of the current flow. Fig. 3 shows the equipotential lines and the lines of equal flow (electrical) in a strip of constant width and thickness. The voltage impressed on the strip is kept constant. An experimental determination of the equipotential lines can easily be made, and the equal flow lines can be drawn in by specifying that a certain part, say one-fifth, of the total electrical flux must flow in each strip of metal contained between adjacent flow lines. Note that by this specification the two boundaries of the strip must be regarded as being flow lines. By choosing the distance between adjacent equipotential lines equal to the distance between adjacent flow lines, the strip is divided into a system of equal squares. Let us speak about these

squares as being cells of equal conductance, a property following directly from Ohm's law.

17 The next step is to apply these elementary considerations to a strip of metal of constant thickness but of variable width. The distribution of the two sets of lines will be as shown in Fig. 4. In the neighborhood of the place where the width changes, the

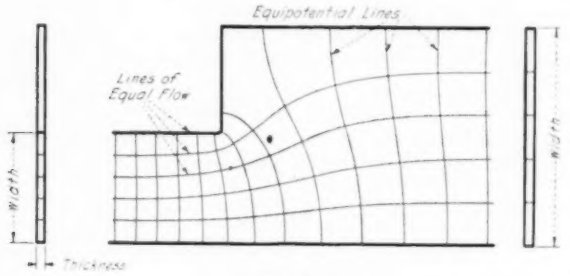

FIG. 4

equipotential lines will become curved, and so will the lines of equal flux since they have been drawn from the consideration of equal-conductance cells. The equipotential lines shown in Fig. 4 have been obtained experimentally.

18 Now consider the flow of electricity in a strip of constant width but of variable thickness. Assume that the thickness of the

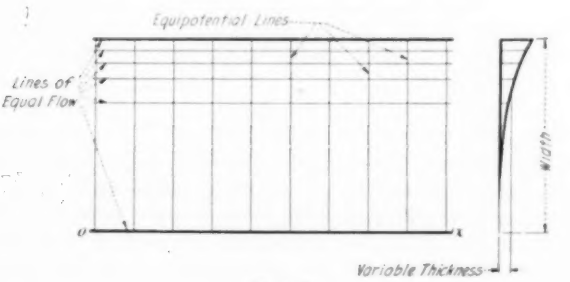

FIG. 5

strip does not vary with the length, but that it varies as the cube of the width and that the variation is gradual. The shape of the cross-section of the strip will therefore resemble the shape of the cross-section of a hollow-ground razor blade. Fig. 5 shows the strip with the distribution of equipotential lines and lines of equal flux. It is seen in this case that the distances between the lines of equal flux will decrease in the direction of the thick side of the strip, while the equipotential lines will continue to lie equal dis-

tances apart. The strip has now been divided into a set of cells, each cell obeying the law expressed by Equation [4], namely, that the length of a cell divided by the width of the same cell is proportional to the cube of the distance the cell is away from the line *ox*. In other words, the equipotential lines become analogous to the equiangular lines, and the lines of equal flux become analogous to the equimomental lines of the torsion problem. The line *ox* is then to be regarded as being the axis of a shaft, while the outline of the thick side of the strip corresponds to the axial outline of the shaft. Thus the strip of constant width and of cubical cross-section will give information about the torsional properties of a shaft of constant diameter, equivalent to Coulomb's simple torsion analysis.

19 Fig. 6 shows a strip of variable width and of a thickness varying as the third power of the width. If the maximum thickness of the strip be small as compared to the width and to the

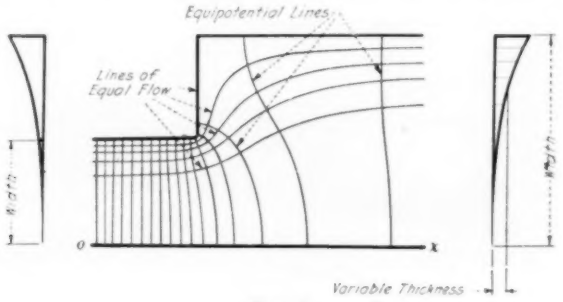

FIG. 6

length of the strip, and if no abrupt variation of thickness occurs, it is justifiable to assume that the distribution of the equipotential lines and lines of equal flux still obeys the law expressed by Equation [4]. Fig. 6 therefore gives the desired layout of the equiangular lines of the torsion analysis of a shaft of the same axial outline as the outline of the metal strip conductor. It will be remembered that Equation [7] gives the stress ratio at any two points of a shaft in terms of the ratio of the distances between the equiangular lines in the neighborhood of the points and the ratio of the radial distances the points are away from the axis. From the electrical model can be obtained the ratio of the distances between the equipotential lines in the neighborhood of the two points by simply measuring the voltage gradients at the two points in question.

EXPERIMENTAL WORK

20 A series of experimental determinations of the electric flow in " razor blade " conductors was undertaken to test the feasibility

of employing the electrical analogy for the solution of torsion problems similar to the one considered above. To accomplish this the model shown in Fig. 7 was built of machine steel. The reason for selecting machine steel was that a high-resistance material was not at hand in the right size and that the machining properties of such a metal were not considered to be favorable for producing an accurate model.

21 Since it would be a practical impossibility to make the side *ox* of the model thin enough (zero thickness) to apply to the case of a solid shaft, the model was in reality made to apply to a hollow shaft; this meant that the thin side of the model could be made of finite thickness. It is evident that the stress distribution in a hollow shaft of 6 in. outside radius and $\frac{3}{4}$ in. inside radius is for all practical purposes the same as the stress distribution in a solid shaft of 6 in. radius.

22 The proportions of the model were decided upon after a few preliminary experiments on the electric flow in plates of con-

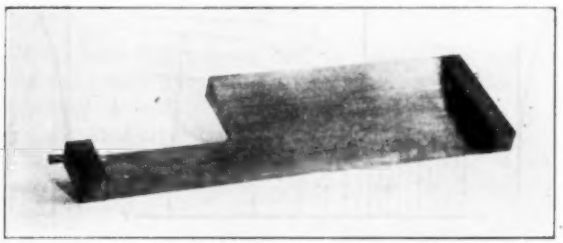

FIG. 7

stant thickness. The experience gained by the making of the models and by the experimental use of them indicates that the thickness could be decreased to about one-half of what was used. As stated before, a thin model, provided it be accurately made, will yield results that are in better agreement with the theoretical than a thick model. This of course is due to the fact that the assumption of a two-dimensional flow is a better approximation for a thin model than for a thick one. The model, or "razor blade," was made 24 in. long, 6 in. wide, and of a maximum thickness of 1 in.

23 The fastening of the copper electrodes to the model caused some difficulty because a very excellent contact was required in order that the current distribution might be uniform. After some experimentation had been made, the method that was adopted was to use silver solder applied by an ordinary blow torch. A test showed that this method gave uniform current distribution.

24 The experiments were carried out in the following way: A constant current (d.c.) was passed through the model. The two

terminals of a very sensitive galvanometer were connected to two sharply pointed needles; the two needlepoints were kept a constant distance apart by having them fastened to a block of fairly rigid rubber. On bringing the two needlepoints in contact with the electrical model, the voltage gradient for the distance between the needlepoints was indicated by a deflection of the galvanometer. Since it was of special interest to obtain the ratio of maximum stress to average stress, the maximum voltage gradients near the fillet and at a point sufficiently removed from the fillet were re-

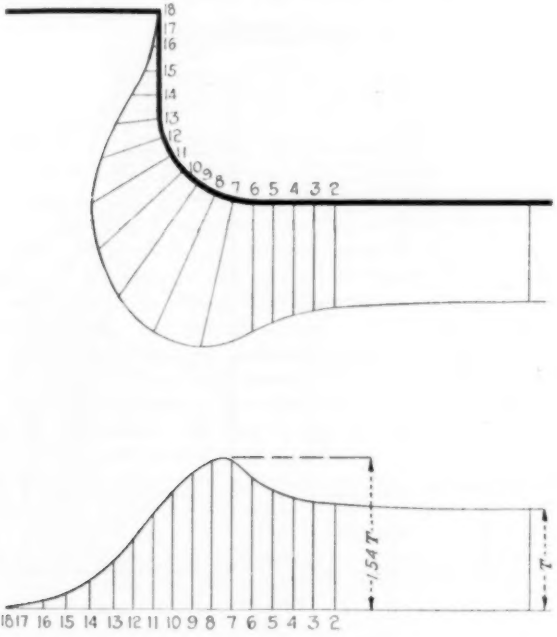

FIG. 8 TORSIONAL-STRESS DISTRIBUTION IN NEIGHBORHOOD OF FILLET

corded. In obtaining the voltage gradient at the fillet care was taken to find the point where the maximum gradient occurred. This necessitated that the distance between the two needlepoints be very small for the exploration of small fillets. In order to decide on how small a distance to use for a certain fillet, a number of consecutive readings along the boundary of the fillet were plotted by using different needlepoint distances; this procedure also located the position of the maximum voltage gradient.

25 Fig. 8 shows the maximum voltage-gradient variation near the fillet. The distance between the needlepoints was in this case one-tenth of the radius of the fillet, or 0.1 in.

26 Due to the fact that the reading of the galvanometer was influenced by stray currents and by thermoelectric effects, a calibration reading was taken with every recorded voltage gradient. It was found necessary to perform such a calibration directly after every reading because the calibration changed continuously. In order to obtain consistent results, the readings were repeated twenty times for each point; ten of these repetitions were carried out on one side of the model, as close to the boundary as possible, and the remaining ten were taken on the other side of the model. Thus for each point shown on the curves at least sixty readings

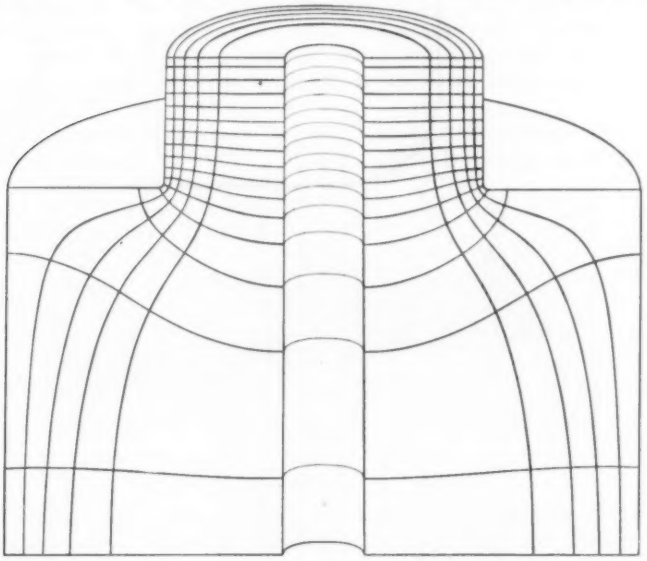

FIG. 9

were recorded, twenty for the maximum gradient at the fillet, twenty for the basis of comparison, and twenty for calibration purposes. The total number of readings for the investigation amounted to more than 2000.

27 After the maximum and the comparison gradients had been found, the maximum torsional stresses were computed by using Equation [7]. Keeping in mind that the voltage gradients are inversely proportional to the distances Δx_1 and Δx_2, Equation [7] is read as follows:

$$\frac{\text{Max. stress}}{\text{Comp. stress}} = \frac{\text{Max. grad.}}{\text{Comp. grad.}} \cdot \frac{\text{Radius to point of max. grad.}}{\text{Radius to point of comp. grad.}}$$

The maximum stress in the shaft of small diameter, sufficiently distant from the fillet, was taken as the basis of comparison.

28 The assumption of a two-dimensional flow is, of course, not correct. Some idea of the inaccuracy due to this assumption may be gained by comparing the maximum voltage gradients obtained on the curved side of the model to those on the straight side of the model. The gradients are those at the fillet. The large width of the model was 6 in.

Small width of model	Fillet radius	Ratio of gradients
$5\frac{1}{2}$	$\frac{1}{2}$	1.01
$5\frac{1}{2}$	$\frac{3}{32}$	1.06
5	1	1.01
5	$\frac{1}{32}$	1.09
$4\frac{1}{2}$	1	1.01
$4\frac{1}{2}$	$\frac{1}{16}$	1.07
4	1	1.02
4	$\frac{1}{16}$	1.09
3	1	1.01
3	$\frac{1}{8}$	1.08

29 It will be seen that the discrepancy increases as the ratio of fillet radius to radius of model width increases; it also increases as the ratio of large width to small width becomes greater. It seems fair, however, to assume that the gradient for a truly two-dimensional flow will lie somewhere between the two measured gradients. If the mean of the two gradients be used, this gives in the most unfavorable case a systematic error that may approach 4 to 5 per cent. The probable error due to faulty contact, to current variation, and to observation has been figured by the method of least squares and amounts to about 1.8 per cent.

30 Generally speaking, the experimental work did not present any great difficulties. As the outline of the model had to be changed before the readings for a point on the curve could be obtained, the time consumed was considerable. In the present investigation 24 different outlines have been studied and recorded on the accompanying curves. Fig. 9 represents the layout of the equiangular surfaces and equimomental tubes as obtained by a very careful tracing of the equipotential lines on the electrical model.

Discussion of Results

31 The results obtained by applying the electrical analogy to the torsion problem of a shaft of two diameters are given in curve form by Figs. 10 and 11. The curves in Fig. 10 show clearly the importance of providing a fillet of adequate size to diminish the stress concentration at the section where the change in diameter occurs. To illustrate the use of the curves, consider a shaft of the two diameters $5\frac{1}{2}$ in. and 4 in. The ratio of these diameters is approximately 1.34, and the stress at the fillet for different fillet sizes is given by curve 3. If a fillet of a $\frac{1}{4}$-in. radius be allowed, the ratio of the fillet radius to the radius of the small shaft will be 0.125. The torsional stress corresponding to this size of fillet is then shown by the curve to be about 1.6 times the maximum torsional

stress in the shaft of smaller diameter. If a fillet of a $\frac{1}{16}$-in. radius had been used, a stress of about 2.3 times the maximum stress in the shaft of smaller diameter would have resulted.

32 Fig. 11 shows the experimental results plotted for several constant ratios of fillet radius to radius of small shaft. It is seen

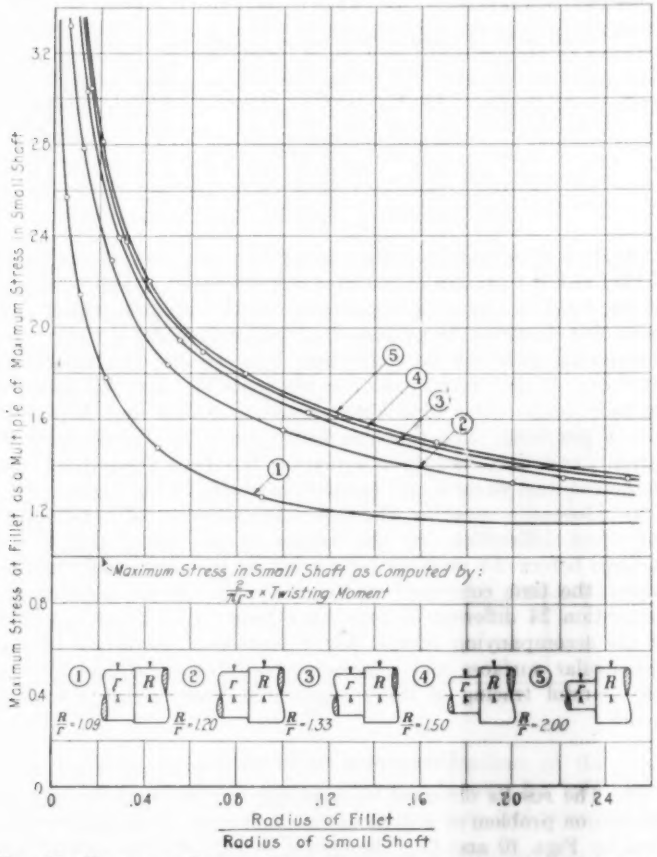

FIG. 10 TORSIONAL-STRESS CONCENTRATIONS IN CIRCULAR SHAFTS OF TWO DIAMETERS

that a variation of the ratio of the two diameters of the shaft does not influence the maximum stress appreciably for values of the diameter ratio greater than two.

33 Fig. 12 shows a comparison of the results published by F. A. Willers[1] and those obtained by the electrical analogy. The agree-

[1] Loc. cit.

ment of the two curves for the constant diameter ratio 1.33 is quite remarkable in spite of the fact that the Willers curve seems to increase at a slightly higher rate than the experimental curve for small values of the radius of the fillet.

34 The agreement of the other set of curves for the 0.10 ratio of fillet radius to radius of small shaft is not very gratifying for the value of $(R-r)/r = 0.10$. The disagreement is not easy to

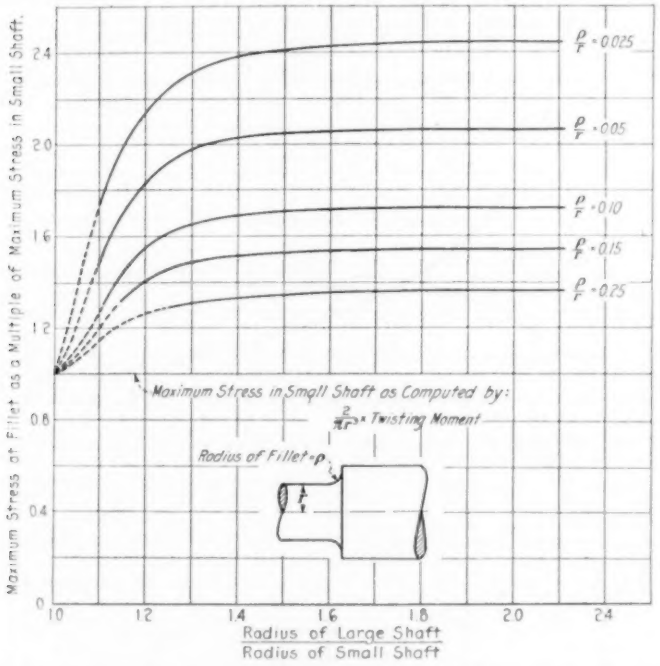

FIG. 11 TORSIONAL-STRESS CONCENTRATIONS IN CIRCULAR SHAFTS OF TWO DIAMETERS

explain, but it is quite improbable that the results obtained by the electrical analogy should be in error at this point, first because the analogy yields consistent results in good agreement with those published by F. A. Willers for the ratio 1.33, and secondly because the point obtained by the electrical analogy for the ratio $(R-r)/r = 0.09$ is a point on a curve that has been drawn through five independently determined points, while the point given by F. A. Willers for the ratio $(R-r)/r = 0.10$ is the result of one graphical determination. On the whole, one is impressed with the check of the results obtained by the two methods.

35 It is hardly necessary to point out the advantages of one method over the other. The inconvenience of having to provide a "razor blade" conductor is more than offset by the saving of time if a series of outlines is to be studied. For a single determination

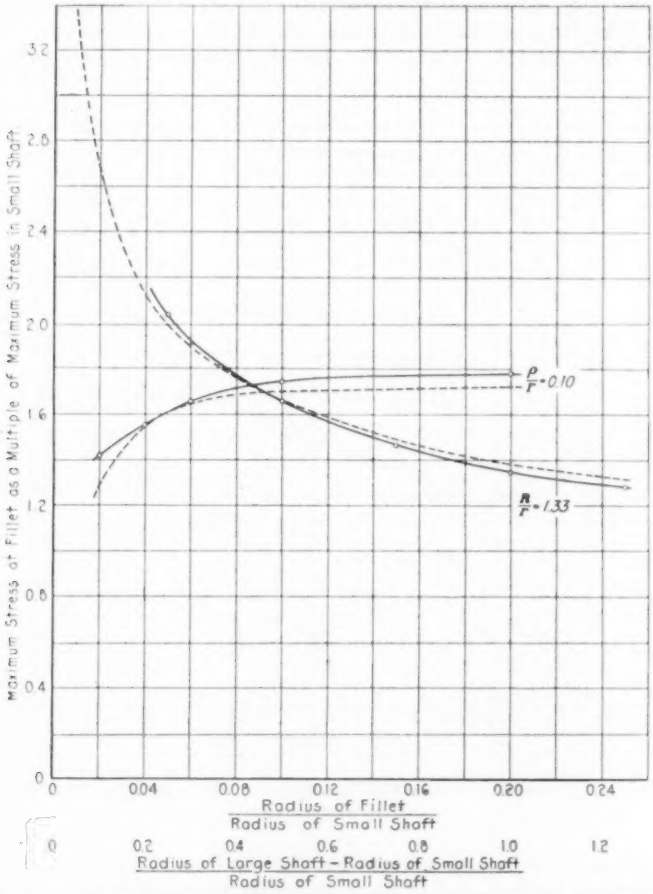

FIG. 12 COMPARISON OF RESULTS OBTAINED BY GRAPHICAL METHOD AND BY ELECTRICAL ANALOGY

The full-line curves have been obtained by means of a graphical method published by F. A. Willers in *Zeit. für Mathematik und Physik*, vol. 55, 1907, pp. 225-263. The dotted-line curves have been obtained by means of the electrical analogy.

of the torsional stresses in a shaft of relatively simple outline it may be found too expensive to provide the necessary electrical model, and the graphical method may be the logical one to which to have recourse. It should be mentioned that torsional stresses in

shafts whose cross-sections are symmetrical with respect to the principal axes may be analyzed with a high degree of accuracy by means of the electrical analogy. Work of this type, relating to torsional stresses in keyways, is being carried out.

36 The investigation reported in this paper was carried out by the author in connection with his regular work in the Motor Engineering Department of the Westinghouse Electric and Manufacturing Company, at East Pittsburgh, while he was still in the employ of that company. The author wishes to acknowledge his indebtedness to Messrs. G. M. Eaton and S. Timoshenko of the Westinghouse Company: to Mr. Eaton for originating and making this investigation possible, and to Dr. Timoshenko for his invaluable encouragement and help in carrying it out.

APPENDIX

TORSION OF A SHAFT OF CIRCULAR CROSS-SECTION AND OF VARIABLE OUTLINE

37 The general solution of this problem was first published by Michell.[1] A few years later, Föppl[2] considered the problem in detail and showed how a hydrodynamic analogy — first suggested by Lord Kelvin[3] — may be applied to give a comprehensive picture of the stress distribution.

38 The following derivation of the differential equation governing the torsional-stress distribution is mainly the same as the one published by Professor Föppl. Instead of following Professor Föppl's derivation of the hydrodynamic analogy, it is shown how an application of Maxwell's equations[4] for the flow of electricity in a conductor of special shape will yield the same differential equation and the same boundary conditions as those of the torsional problem.

39 The coördinate system will be chosen in such a way that the X axis coincides with the axis of revolution of the shaft, the elastic displacements along the coördinate axes being denoted by u, v, and w. The following fundamental equations of the theory of elasticity must then be satisfied:

$$\left.\begin{aligned} \nabla^2 u + \frac{m}{m-2}\frac{\partial e}{\partial x} + \frac{X}{G} &= 0 \\ \nabla^2 v + \frac{m}{m-2}\frac{\partial e}{\partial y} + \frac{Y}{G} &= 0 \\ \nabla^2 w + \frac{m}{m-2}\frac{\partial e}{\partial z} + \frac{Z}{G} &= 0 \end{aligned}\right\} \quad \ldots\ldots\ldots [8]$$

By ∇^2 or Laplace's operator must be understood

$$\nabla^2 = \frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} + \frac{\partial^2}{\partial z^2}$$

[1] J. H. Michell, Proc., Math. Soc., London, vol. 31, 1900, pp. 141-142.

[2] August Föppl, Sitzungsberichten der Kgl. Bayrischen Akademie d. Wissenschaften, vol. 35, 1905, p. 249, and Vorlesungen über Technische Mechanik, vol. 5, pp. 183-202.

[3] Thomson and Tait, Treatise on Natural Philosophy, Part 2, pp. 242-243.

[4] Riemann-Weber, Die Partiellen Differentialgleichungen, 1900 Edition, pp. 429-431.

The symbol e denotes the volume change or dilatation, while X, Y, and Z stand for the body forces along the coördinate axes. The coefficient of lateral contraction is represented by $1/m$, and the modulus of elasticity for shear is denoted by G.

40 Professor Föppl's solution from now on is based on the assumption that cross-sections perpendicular to the axis of the shaft remain plane after the shaft has been given an angle of twist; a new variable, p, is also introduced. The physical significance of p is the angle through which the infinitesimal annulus of width ∂x, radius r, and thickness ∂r, twists when a moment is applied to the shaft. By putting

$$r^2 = y^2 + z^2 \quad \ldots\ldots\ldots\ldots [9]$$

we see that the new variable p must be a function of the radius r and the axial coördinate x. The elastic displacements may now be put down as follows:

$$u = 0; \qquad v = pz; \qquad w = -py \quad \ldots\ldots\ldots [10]$$

By substituting these expressions for the elastic displacements in the expression for the dilatation

$$e = \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z}$$

it is seen that the dilatation in this case becomes zero. Since we deal with stresses that are high compared to those induced by the weight of the shaft material, and since we assume that no external forces act on the shaft in the neighborhood of the region under investigation, we may place the body forces X, Y, and Z equal to zero. The fundamental equations then reduce to the simple form:

$$\nabla^2 u = 0; \qquad \nabla^2 v = 0; \qquad \nabla^2 w = 0 \quad \ldots\ldots\ldots [11]$$

The first of these equations is satisfied at once, since we have already made the assumption that there are no elastic displacements in the x direction, namely, that $u = 0$.

41 By means of Equations [9] and [10], we may obtain the second derivatives of v and w with respect x and r in terms of p. The substitution of these in Equation [11] gives the following differential equation for p, x and r:

$$\frac{\partial^2 p}{\partial x^2} + \frac{\partial^2 p}{\partial r^2} + \frac{3}{r} \cdot \frac{\partial p}{\partial r} = 0 \quad \ldots\ldots\ldots [12]$$

42 The problem is now to find the function that will satisfy this equation and the geometric boundary conditions of the shaft. The boundary conditions require that the direction of torsional stress in an axial plane at the boundary coincide with the tangent of the outline of the shaft at that point. The torsional-stress components T_{yz} and T_{yx} must satisfy the following fundamental equations of the theory of elasticity:

$$\left.\begin{aligned} T_{yz} &= G\left[\frac{\partial v}{\partial z} + \frac{\partial w}{\partial y}\right] \\ T_{yx} &= G\left[\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right] \end{aligned}\right\} \quad \ldots\ldots\ldots [13]$$

Expressing these components in terms of p, x, and r, and remembering that we may consider an axial plane where $y = 0$ and $z = r$, we obtain for the torsional-stress components:

$$\left.\begin{aligned} T_{yz} &= G\,\frac{z^2 - r^2}{r}\,\frac{\partial p}{\partial r} = Gr\,\frac{\partial p}{\partial r} \\ T_{yx} &= Gz\,\frac{\partial p}{\partial x} = Gr\,\frac{\partial p}{\partial x} \end{aligned}\right\} \quad \ldots\ldots\ldots [14]$$

The boundary conditions are now expressed by introducing the differentials dx and dr of the geometric outline of the shaft in the axial plane. The expression for the boundary conditions becomes

$$\frac{dr}{dx} = \frac{T_{yz}}{T_{yx}} = \frac{\dfrac{\partial p}{\partial r}}{\dfrac{\partial p}{\partial x}} \quad \ldots\ldots\ldots\ldots [15]$$

43 The differential equation of the function p and the boundary conditions that p must satisfy have now been obtained. From the definition of p it follows that

$$p = c_1 = \text{a constant}$$

will represent the trace of a surface on the axial plane $y = 0$ and $z = r$, the surface containing all the points that have rotated through a certain angle of twist with respect to the points lying on another surface whose trace in the axial plane is given by the equation

$$p = c_2 = \text{a constant.}$$

The equation of an equiangular surface — see definition in qualitative analysis — will therefore be

$$p = c = \text{a constant} \quad \ldots\ldots\ldots\ldots [16]$$

44 It will now be shown that the function p of the torsional problem is analogous to the potential function of an electrical problem.

45 In a chapter on the steady state of electric-current flow in thin plates, Professor Weber[1] reduces the fundamental differential equations of Maxwell to the following:

$$\frac{\partial\left(\lambda h \dfrac{\partial \phi}{\partial x}\right)}{\partial x} + \frac{\partial\left(\lambda h \dfrac{\partial \phi}{\partial y}\right)}{\partial y} = 0 \quad \ldots\ldots\ldots [17]$$

where ϕ is the potential function, h is the thickness of the plate, and λ the conductivity constant of the material. The thickness must be small compared to the other dimensions of the plate, but it is *not* necessarily constant.

46 Consider a plate of the "razor-blade" type, i.e., a plate where

$$h = \alpha y^3 \quad \text{and} \quad \lambda = \text{a constant} \quad \ldots\ldots\ldots [18]$$

Substituting this expression into [17], we get an equation for the potential function, ϕ, corresponding to the electric-current flow in the "razor-blade" conductor:

$$\frac{\partial^2 \phi}{\partial x^2} + \frac{\partial^2 \phi}{\partial y^2} + \frac{3}{y}\frac{\partial \phi}{\partial y} = 0 \quad \ldots\ldots\ldots [19]$$

It is seen that Equations [12] and [19] are analogous. In other words, the trace of the equiangular surface in the axial plane for

$$p = c = \text{a constant}$$

is analogous to the trace of the equipotential surface on the side of the "razor blade" for the corresponding expression

$$\phi = c = \text{a constant}$$

[1] See footnote 4 on page 635.

47 The boundary conditions of the electrical problem require that the equipotential surfaces intersect the boundary of the "razor-blade"

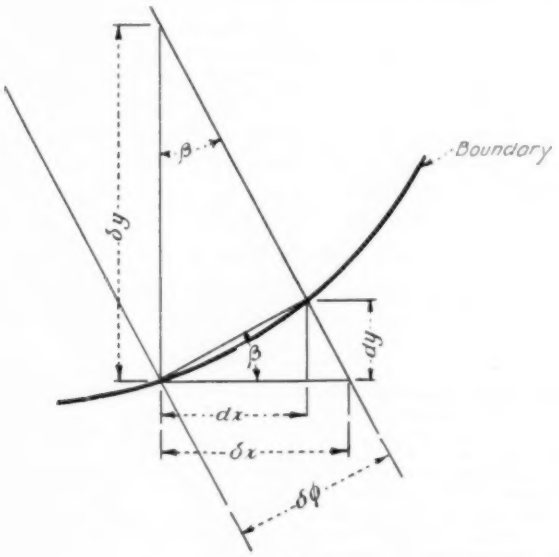

FIG. 13

conductor at right angles. Referring to Fig. 13, it is seen that the geometrical slope of the boundary is

$$\frac{dy}{dx} = \tan\beta \quad \ldots\ldots\ldots\ldots \quad [20]$$

Furthermore

$$\sin\beta = \frac{\partial\phi}{\partial y}\,; \quad \cos\beta = \frac{\partial\phi}{\partial x} \quad \ldots\ldots\ldots \quad [21]$$

Combining [20] and [21], we get an expression analogous to Equation [15], thereby establishing the completeness of the analogy of the torsional problem and the electrical problem

$$\frac{dy}{dx} = \frac{\dfrac{\partial\phi}{\partial y}}{\dfrac{\partial\phi}{\partial x}} \quad \ldots\ldots\ldots\ldots \quad [22]$$

DISCUSSION

S. Timoshenko.[1] Mr. Jacobsen's paper is very interesting and one of great practical importance. It is well known that sharp variations in cross-section, reëntrant corners, and fillets of small radii produce very high stress concentration. These high local stresses, although not dangerous in the case of constant loading and ductile material, may produce a pronounced weakening effect in the case of stress fluctuation, which usually takes place in machine parts. Many cases of fractures in service can be attributed to this cause.

It is only for the simplest cases that the problem of stress concentration can be solved analytically. For the more difficult cases one of a number of experimental methods must be applied in studying the local stresses. In the case of two-dimensional problems the photoelastic method and the Lüders-line method have proved very satisfactory. These methods cannot be applied in the case of torsion, and all our knowledge on stress concentration produced at the fillets of shafts under torsion has been obtained by an approximate analytical solution. This solution, although accurate enough, involves much work and cannot be easily applied to practical cases. Mr. Jacobsen has shown in his paper how the torsional problem can be solved experimentally by using the analogy between the torsion problem and that of the flow of electricity in plates of variable thickness.

In our photoelastic work we use celluloid models. Some experiments with bakelite have also been made in our laboratory. The most important difficulty in experimenting with these materials is the "edge effect." We try usually to complete the work on a model in one day, i.e., before any noticeable edge effect appears. By taking several readings at different loads and plotting these readings against the values of loads, a straight line will result. From the angle of inclination of this line the true value of the stress produced by the loading may be obtained. In this manner the edge effect is eliminated from our calculations. The same method can be used also in the case of bakelite, which usually, however, is less satisfactory than celluloid with respect to homogeneity.

Paul A. Heymans.[2] Mr. Jacobsen has developed for the problem which he has investigated, ingenious similarity conditions between an electrical model and his problem of stress distribution.

[1] Research Engineer, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa. Mem. A.S.M.E.

[2] Assistant Professor of Theoretical Physics and Photoelasticity, Massachusetts Institute of Technology, Cambridge, Mass. Assoc-Mem. A.S.M.E.

One cannot but regret that the very principles of this skilful method greatly limit its field of application.

Indeed, it has been possible for Mr. Jacobsen to obtain a similar electrical model in this particular case dealing with torsional stresses in shafts of circular cross-section on account of the existence of the conditions of similarity expressed by Equations [**4**], [**5**], [**6**], and [**7**] of his paper. Such relations, however, will exist only in a restricted class of stress problems. In fact, the method seems to be confined to one-dimensional-stress problems and therefore — although being of great interest for this class of problems — it seems at present of limited interest for the solution of problems of stress distribution in general.

As Dr. Timoshenko mentioned the use of bakelite for photoelastic analysis, the writer might state that Mr. Frost and himself have made a special investigation of that material and found[1] that some examples of commercial bakelite were fairly free from internal stress, and that their optical properties and their small elastic and optical hysteresis made bakelite a better material than celluloid for use in the photoelastic method. Removing the internal stresses in bakelite by heat treatment has also been attempted, and has been fairly successful in a number of cases.

The method which consists of determining the increment of stress due to a known increment of load and thereby removing the difficulty of permanent initial stresses, is not always valid. Indeed, the pattern of isoclinic lines may be distorted by the initial stresses; also the magnitude of the initial stresses may be beyond the limit of linear proportionality between stress and strain. This limit being lower for celluloid than for bakelite, represents an additional advantage in favor of bakelite.

CLOSURE TO DISCUSSION.[2] Coming now to the discussion of the paper, and in reply to the question of Professor Heymans, it must be noted that Jacobsen's method can be applied in all cases where the differential equation is identical with that of the torsional problem. It can be used, for instance, in the solution of certain heat-flow problems. The two-dimensional problems of elasticity involve the solution of a partial differential equation of the fourth order, and we cannot expect satisfactory results by applying the electrical model in this case. These problems can very easily be solved by using the photoelastic method.

Regarding the application of the electric analogy in the case of combined bending and twist of circular shafts of variable section, it must be noted that the case of combined stress can always

[1] Paul Heymans and T. H. Frost, Stress-Strain Properties of Bakelite and the Law of Its Optical Behavior under Mechanical Stress, *Physical Review*, vol. 25, p. 900.

[2] Closure made by Dr. Timoshenko, who presented the paper for Dr. Jacobsen.

be separated into simpler problems and solved by using the principle of superposition. We can separately solve the problem of bending and of twist of shafts, and obtain the final result by combining the stresses for these two cases in the usual manner. The problem of bending of shafts of variable section involves the solution of more complicated equations than those of twist, and this problem has not yet been solved analytically. In order to get some approximate solution of this problem for the case of stress concentration produced by fillets, we can use two-dimensional models and the photoelastic method. It can be assumed that in the cases where the radius of the fillet is small in comparison with the radius of the shaft the results obtained on such a two-dimensional model represent a good approximation to the three-dimensional problem.[1]

THE AUTHOR. The remark by Professor Heymans that the electric "method seems to be confined to the solution of one-dimensional-stress problems" requires modification.

The problem of torsion is a three-dimensional problem, but if axial symmetry is present it reduces to a pseudo two-dimensional problem. A further reduction of the problem to one dimension necessitates that the bar subjected to torsion be of constant diameter, and for this simple case Coulomb found the theoretical solution prior to the year 1784. It should be pointed out that the photoelastic method of investigation is limited to special two-dimensional problems; moreover, that the photoelastic method is not capable of solving the pseudo two-dimensional problem of torsion.

The electrical method is theoretically valid for solving the problem of torsion considered in this paper. The application of this method to problems not coming within the class of the one dealt with here will require careful analysis to determine whether or not the partial differential equations for the torsional-stress distribution can be made analogous to the equations for the electric current distribution. The electrical method applied to new problems is therefore not foolproof in the hands of a person who is without an appreciation of the fundamental equations of the theory of elasticity as well as with Maxwell's theory of electricity.

It is useless to argue about the relative importance of a crankshaft and a connecting rod when both are indispensable. Similarly, the relative importance of torsion problems and of special two-dimensional-stress problems cannot be evaluated; they are both necessary in order to guide the designer intelligently.

[1] See paper by the writer and W. Dietz: Stress Concentration Produced by Holes and Fillets, p. 199, ante.

No. 1975

TANGENTIAL VIBRATION OF STEAM-TURBINE BUCKETS

By Wilfred Campbell [1]

At the spring meeting of this Society in 1924 a paper was presented by the author on The Protection of Steam-Turbine Disk Wheels from Axial Vibration.[2] *The present paper describes how the same research was extended to include tangential vibration. Substantially the same testing apparatus was used and the same methods of protection have been adopted. Resonant conditions are avoided by carefully prescribed margins. In unsymmetrical reaction buckets, the component vibration lying most nearly in the plane of the wheel is, by definition, treated as tangential vibration. The methods described suffice for protection against any combined or intermediate type. Additional results are given by W. C. Heckman in Part II.*

SHORTLY after the discovery that the stationary wave was the most important single cause of axial-vibration failures, several peculiar cases of bucket breakages occurred on 35,000-kw. turbines. Three of the wheels affected were very similar in physical dimensions. All of these wheels had been carefully designed and tested to insure that critical speeds for axial vibration were a satisfactory distance from the rated running speeds. In order to accomplish this purpose somewhat lighter buckets were used than had been the previous practice on these large wheels.

2 In the case of the first trouble, it was thought that the buckets broke owing to having been injured previously, as these buckets had been run in proximity to a 4-node critical speed for a considerable time and had been replaced after having been reduced in section. This is now known not to have been the

[1] Wilfred Campbell died at Schenectady, N. Y., July 7, 1924. Before his death he had intended to present this paper, which is composed of letters and memoranda in his handwriting or dictated by him. Few alterations have been made other than those required to fit the fragments into a connected story.

[2] Trans. A.S.M.E., vol. 46 (1924), p. 31.

Presented at the Annual Meeting, New York, November 30 to December 4, 1925, of The American Society of Mechanical Engineers.

trouble, the real difficulty being tangential vibration of the first type similar to what is shown in Fig. 1. Figs. 2 and 3 illustrate tangential vibrations of the third and fifth types, respectively.

3 At the next opportunity a mechanical vibrator was taken out and the frequencies of the nodal systems as taken in the factory were checked. In addition, records of the tangential vibration frequencies of various buckets on the wheels were also made.

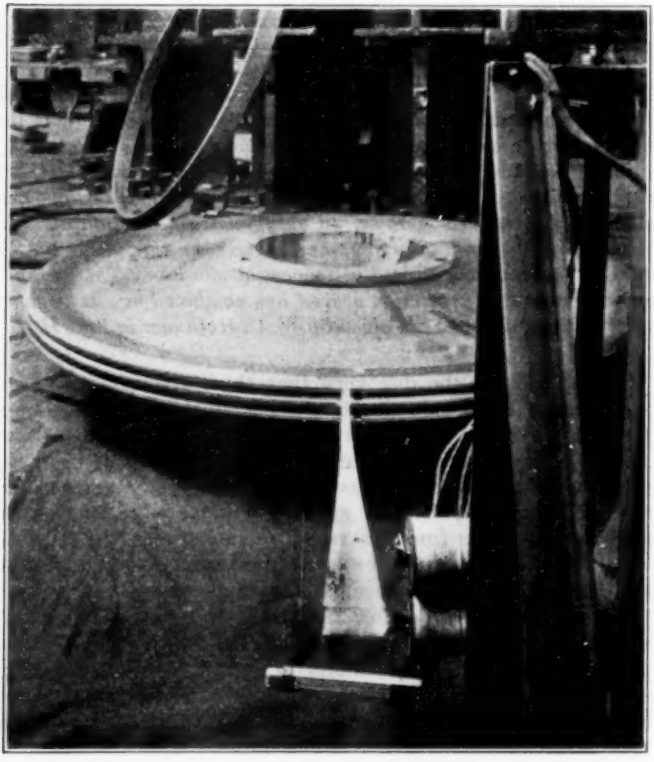

FIG. 1 FIRST TYPE OF TURBINE-BUCKET VIBRATION

These varied from 32 to 33½ cycles per second in different places on the wheel for the first type.

4 From an approximate analysis it was expected that this frequency would rise, due to centrifugal force, to about 50 cycles per second, which was twice the running speed of the machine. Thus there would be two natural vibration cycles of the bucket in a tangential direction for one revolution of the turbine.

5 The fundamental formula of vibration gives f the frequency in cycles per second as:

$$f = \frac{1}{2\pi}\sqrt{\frac{F}{M}} \quad \ldots\ldots\ldots\ldots \quad [1]$$

where

F = the stiffness of the member in lb. per ft. deflection
M = the mass of the vibrating body in slugs = w/g

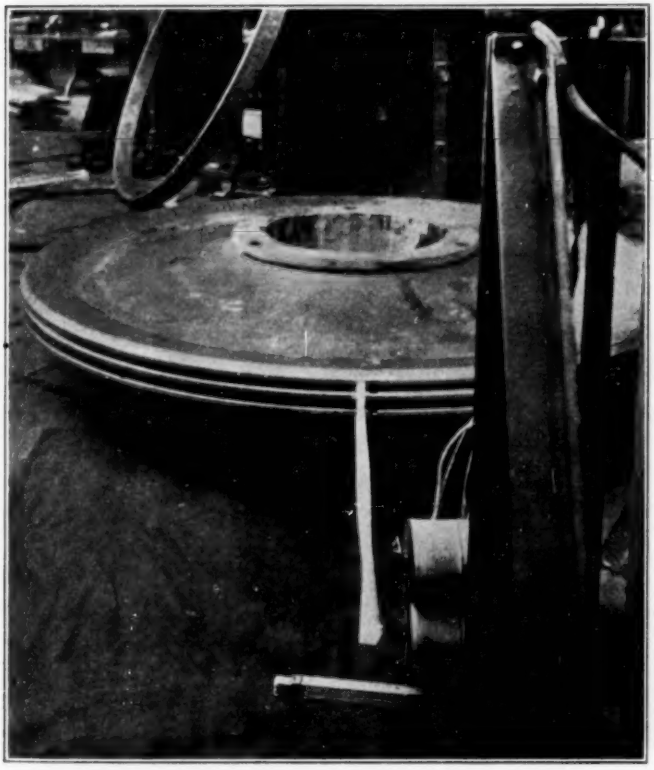

FIG. 2 THIRD TYPE OF TURBINE-BUCKET VIBRATION

6 Since F, the stiffness, is increased by the centrifugal forces tending to cause the bucket to assume a perfect radial position, it is evident that f, the frequency, also is increased by the centrifugal forces.

7 A test in the wheel-testing machine (see author's paper on Axial Vibration) of a similar wheel, except that it has a bucket

of the dimensions used previously to the present trouble, shows that the standing frequency was 35.6 per sec. Tangential vibrations were registered as follows:

4 cycles per revolution at 575 r.p.m.
3 cycles per revolution at 860 r.p.m.
2 cycles per revolution at 1680 r.p.m.

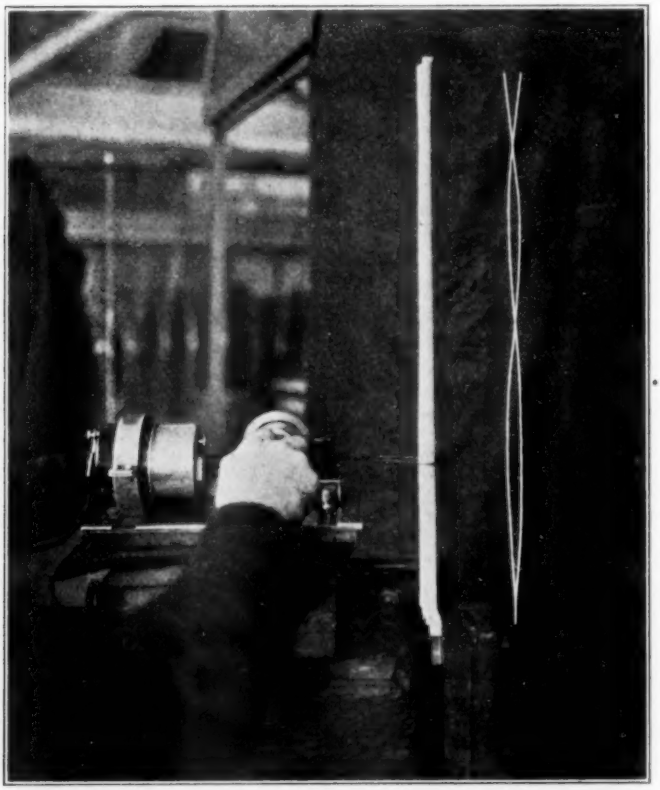

FIG. 3 FIFTH TYPE OF TURBINE-BUCKET VIBRATION

Thus at 1500 r.p.m. no tangential vibration existed (see Fig. 4). The coefficient for change of frequency due to centrifugal force obtained on the wheel tested confirmed that found by calculations and used in the case of the broken buckets previously described.

8 Another wheel had also been run similar to the next to the last stage of one of the above turbines, which also lost buckets. In this case the tangential frequency, when running, of three cycles

per revolution occurred at a speed of 1554 r.p.m. A little lower frequency would cause this to equal 1500 r.p.m. A bucket of higher tangential frequency was designed to replace this.

9 These running vibrations were recorded by means of a coil, as in Fig. 5, carried on a solid wheel and so placed as to register the tangential motion of a small projection placed on a section

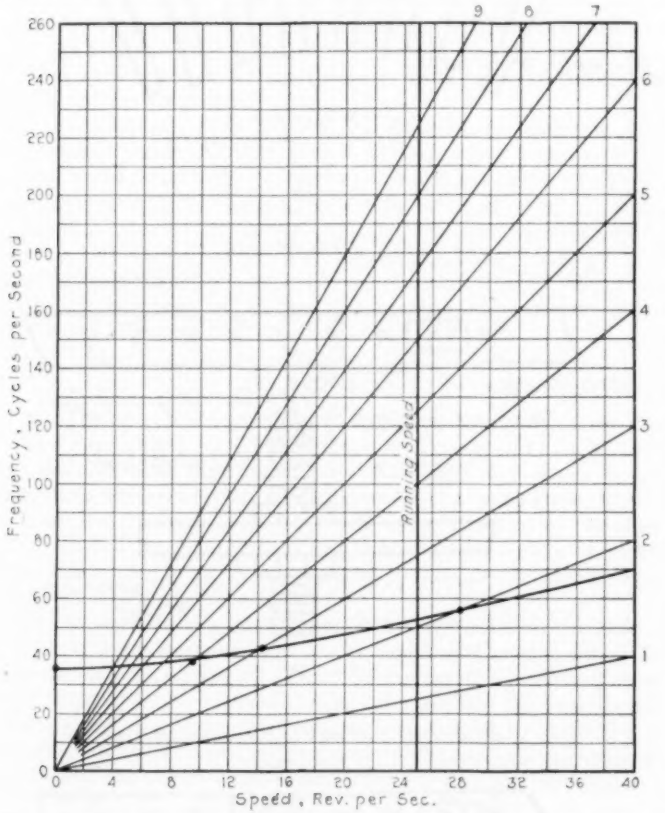

FIG. 4 FREQUENCY-SPEED DIAGRAM SHOWING FIRST TYPE OF TANGENTIAL VIBRATION FOR AN OLD DESIGN OF BUCKET WHICH GAVE NO TROUBLE

of the buckets. An impulse of very brief duration at each revolution could be given by means of a small steam jet striking a small projection attached to the wheel's circumference. This added stimulus was not absolutely necessary for exciting the vibration, the only requirement being that the natural frequency of tangential vibration of the bucket be an exact multiple, for example, 2,

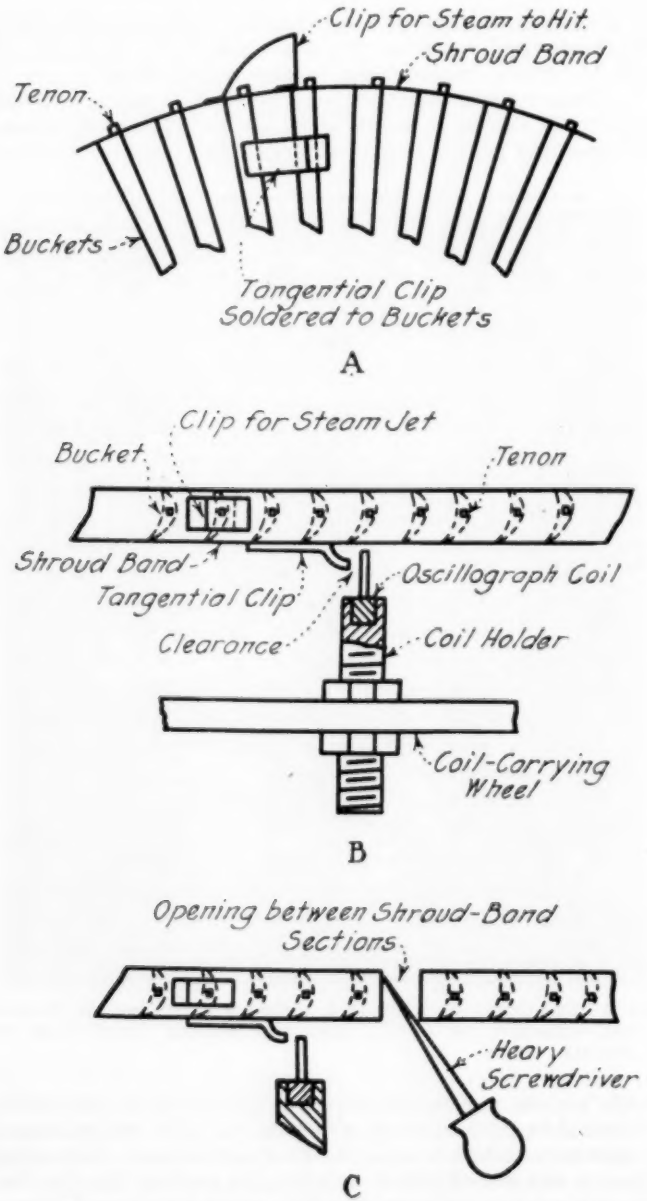

FIG. 5 DEVICES FOR RECORDING TANGENTIAL VIBRATION
(A — Side view; B — Plan view; C — Method of taking standing frequencies.)

3, 4, etc., of the speed of revolution. Smaller buckets are not expected to give this kind of trouble, as their multiples are found to be very much higher.

10 From the knowledge thus obtained, it was shown that long and slender buckets could be safely used, provided that their tangential vibration frequencies do not have resonance at operating speed and that axial wave phenomena be guarded against. The laws governing vibration of the buckets on the disks in a direction at right angles to the shaft were thus outlined, and means were devised to determine, on the rotating wheel, the several critical speeds for this form of vibration.

11 The remedy for tangential vibration lies in *designing* and, if necessary, *tuning* to obtain such frequencies as to prevent the possibility of a critical speed within prescribed limits of normal operating speed. By vibrating a bucket before entering it on the wheel it is possible by tuning to insure freedom from tangential vibration at normal speed. The knowledge gained by these later investigations explains why it is possible satisfactorily to run the longer buckets, and confidence is assured for the safe use in the future of still longer buckets.

Tuning Buckets

12 The two chief properties that control the value of the frequency are as shown by the fundamental Formula [**1**] to be, first, the stiffness, and second, the mass. Like all other elastic bodies the frequency is directly proportional to the square root of the stiffness and inversely proportional to the square root of the mass.

13 It follows, therefore, that the natural frequency of vibration of a bucket row can be changed by any means which varies either the stiffness or the mass.

14 In the case of a bucket row of the type wherein the ends of the buckets are fastened together by a sectional bucket cover, the stiffness of the bucket row may be increased by making more rigid the connection between the bucket cover and the bucket ends. The rigidity of this connection can be increased by soldering or brazing the bucket cover to the bucket ends, and this increases the stiffness of the bucket row and hence increases its natural frequency of vibration.

15 Again, the stiffness of the buckets may be increased by providing circumferentially extending stiffening members between the roots and the tips of the buckets. This sort of tuning is particularly effective with higher types.

16 The natural frequency of vibration of a bucket row can be increased by decreasing the mass of the buckets, care being taken to remove metal at a point where its removal will affect the stiffness by the least amount; i.e., at the outer ends of the buckets.

17 To decrease the natural frequency of vibration of buckets, the most convenient method is to decrease the stiffness by remov-

ing metal from the backs of the buckets, particularly in the region of their inner ends.

Determination of Bucket Dimensions for a Predetermined Frequency by Similitude

18 In the author's paper on axial vibration the formula for the frequency of vibration of a reed was given as follows:

$$f = \frac{t}{l^2} C \quad \ldots\ldots\ldots\ldots [2]$$

where

f = the frequency, cycles per sec.
l = length, in.
t = thickness, in.
C = constant of proportionality.

19 In a turbine bucket this formula may be applied by using a factor S to represent the scale of equivalent thickness, where S is determined from a particular design of bucket by the relation

$$S = \frac{fl^2}{C}$$

20 Then, for similar designs of bucket,

$$C = \frac{f_1 l_1^2}{S_1} = \frac{f_2 l_2^2}{S_2}$$

Therefore the ratio of scales is

$$\frac{S_2}{S_1} = \frac{f_2 l_2^2}{f_1 l_1^2} \quad \ldots\ldots\ldots\ldots [3]$$

For example, a sample bucket 14 in. long has a frequency of 62. What will be the scale for a similar bucket 24 in. long for a desired frequency of 37?

$$\frac{S_2}{S_1} = \frac{37 \times 24^2}{62 \times 14^2} = 1.755$$

21 The required thickness of the new bucket is therefore 1.755 times the thickness of the sample tested.

Speed Coefficient

22 All the experimental information shows that there is a different centrifugal speed coefficient for the case of tangential vibration than for that of axial vibration.

23 The speed coefficient B is determined from the formula

$$f_r = \sqrt{f_s^2 + BN_s^2}$$

where

f_r = running frequency recorded in the wheel-testing machine
f_s = standing frequency
N_s = speed in revolutions per second at which f_r is recorded.

24 Fig. 6 is a chart made from wheel-test results in the wheel-testing machine. The diagonal lines represent multiples of the speed of rotation, the lowest one being 1 per revolution, the next one 2 per revolution, the next 3 per revolution, and so on.

25 The tangential frequency when standing is shown at 78 cycles; at a speed of 13.65, 6 cycles per revolution were recorded.

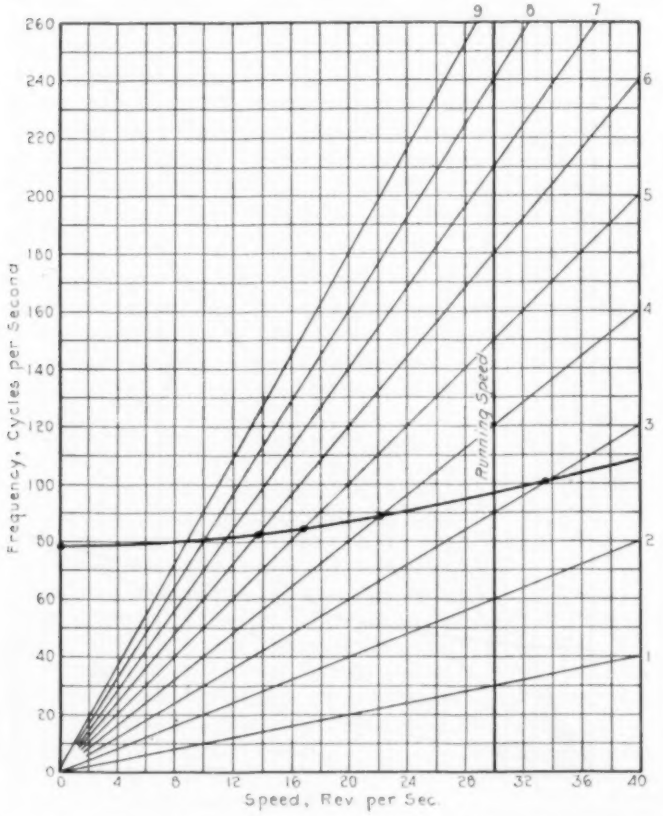

FIG. 6 FREQUENCY-SPEED DIAGRAM FOR FIRST TYPE OF TANGENTIAL VIBRATION

The running frequency also recorded at this speed was 82.25. This crosses the 6-per-revolution diagonal line. Similar remarks serve for the other points along these lines.

26 These frequencies are recorded by means of a coil-carrier wheel carrying a coil in such a manner as to register any change in velocity due to the vibration of the buckets.

27 We have had buckets break when intersections of the No. 2 and 3 lines occur at the running-speed line. We are not so sure of any having occurred at the 4-per-revolution or beyond.

28 A common practice in Europe and elsewhere has been to avoid entirely the use of buckets having a standing frequency of less than 3.5 times the running speed. This would limit long

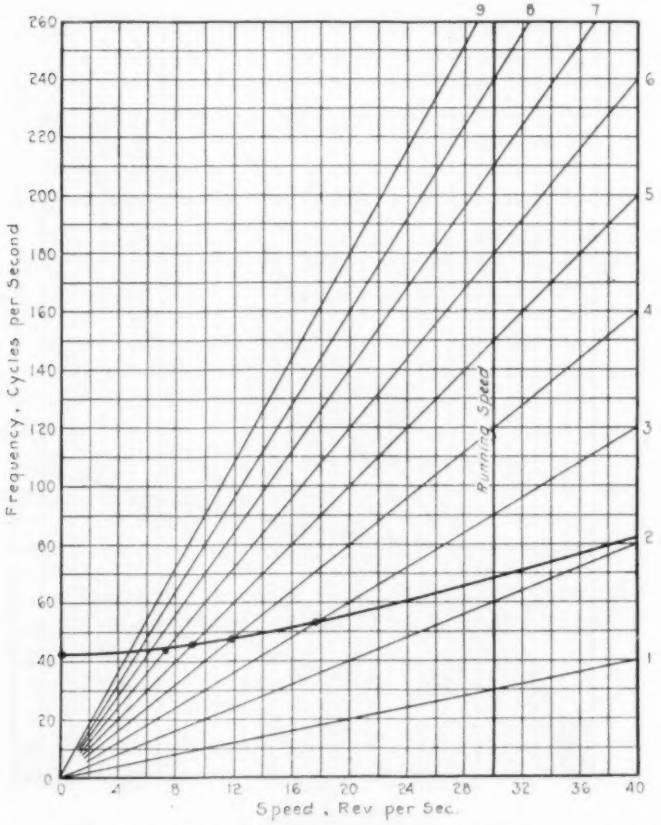

FIG. 7 FREQUENCY-SPEED DIAGRAM FOR FIRST TYPE OF TANGENTIAL VIBRATION, SHOWING SATISFACTORY MARGINS

buckets to about one-half the lengths which have been found successful; but as we have been using for years hundreds of buckets that come within the 1: 3.5 ratio, we now understand the specific requirements for their safety.

29 Fig. 7 shows wheel-test results taken on a turbine wheel at Schenectady. The curve represents tangential frequency of

the buckets at the various speeds of the machine, standing frequency being 42 cycles per sec. Frequencies at all of the speeds can be found from the formula

$$f_r = \sqrt{f_s^2 + BN_s^2}$$

The speed coefficient B in this case has been obtained from the test data. The radiating lines from the corner marked 0 represent impulses of 1 per revolution, 2 per revolution, 3 per revolution, 4 per revolution, 5 per revolution, etc.

30 It will be noted that the curve crosses one of the radiating lines at a speed of 7.3 revolutions per second. This means that 6 cycles per revolution were recorded at this speed. Likewise, 5 is recorded at 9.1 revolutions per second; 4 per revolution at 11.8; and 3 per revolution at 17.6. It is expected that 2 per revolution would occur at some point beyond 40 revolutions per second. This point was not recorded owing to the wheel's not having been run at this extremely high speed. The other points were recorded by means of the rotating coil in the wheel-testing machine, acted on by a small projection silver-soldered to a bucket.

31 We have found that the most hazardous condition would be at the intersection of the 2-per-revolution line. Therefore we attempt to design and tune buckets so that the 2-per-revolution point will be at least 15 per cent from the running speed; the 3-per-revolution at least 10 per cent, and the 4-per-revolution at least 5 per cent. Usually the amplitudes of motions for the 5, 6, and 7 points are so small that we ignore their location. We do not find much trouble in buckets below 10 in. long, but if these are exceeded, then some check should be made on each type of bucket.

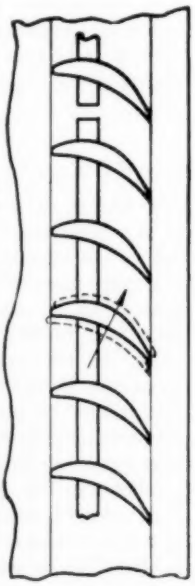

FIG. 8 UNSYMMETRICAL REACTION BUCKETS. THE MOTION ILLUSTRATED IS DEFINED AS TANGENTIAL VIBRATION

REACTION BUCKETS

32 In the case of reaction buckets, or unsymmetrical blades whose axes of vibration do not lie exactly axial and tangential, it was noted in stationary tests that the vibratory motion was in a diagonal direction, this being at right angles to the axis of the minimum inertia of the blade shape. See Fig. 8. This type of vibration, which for reaction blades is defined as tangential vibration, seems to be the cause of far more trouble than the corresponding case of axial vibration of buckets.

33 As in the case with symmetrical buckets, in order to determine resonance conditions at normal running speed, the effects of centrifugal force on the frequencies recorded under stationary conditions have to be calculated.

PART II

By W. C. Heckman [1]

MR. CAMPBELL'S untimely death while in the midst of preparing a paper on bucket vibration left the paper brief and incomplete. Since this paper covers work and observations made by the group of which he was leader, it is only fitting at this time to add those results and observations which are rightfully a part of the paper.

Types of Bucket Vibration

35 If a long turbine bucket is clamped in a vise with one end free and an exciting force of an increasing frequency is applied to it in a tangential direction, a certain value of frequency is reached where the bucket vibrates back and forth in the same manner as if it were plucked and allowed to vibrate freely until it returned to rest. This is called the fundamental frequency or first type of tangential bucket vibration. Fig. 1 illustrates this.

36 As the applied frequency is increased another point of resonance is reached where a node is established at approximately two-thirds the length of the bucket from the clamp. The free end vibrates back and forth. This is defined as the *third* type of vibration since the figure described is similar to a three-quarters wave as shown in Fig. 2.

37 Again increasing the frequency another point of resonance is reached where two nodes are established and the free end vibrates. This is defined as the *fifth* type since the figure is similar to a five-quarters wave, and so on. The fifth type is illustrated in Fig. 3.

38 If the outer end of the bucket as well as the root is also held in a clamp and the above process repeated, a point of resonance is reached at a frequency between the first and third types where the bucket vibrates with its maximum amplitude near the center of its length. This is defined as the second type since it approximates a half-wave or two quarters.

39 At a higher frequency, between the third and fifth types, a point of resonance is reached at which a node is established approximately half-way between the two clamps. This is defined as the fourth type.

[1] Turbine Engineering Department, General Electric Co.

40 At a frequency between the fifth and seventh types the sixth type occurs where two nodes are established in the bucket between the clamps. And so on.

41 Conditions for the establishment of the even-numbered types are provided in turbine-wheel and bucket construction by the lashing or shroud bands at the outer ends of the bucket. In order that the outer ends of the buckets remain fixed relatively to

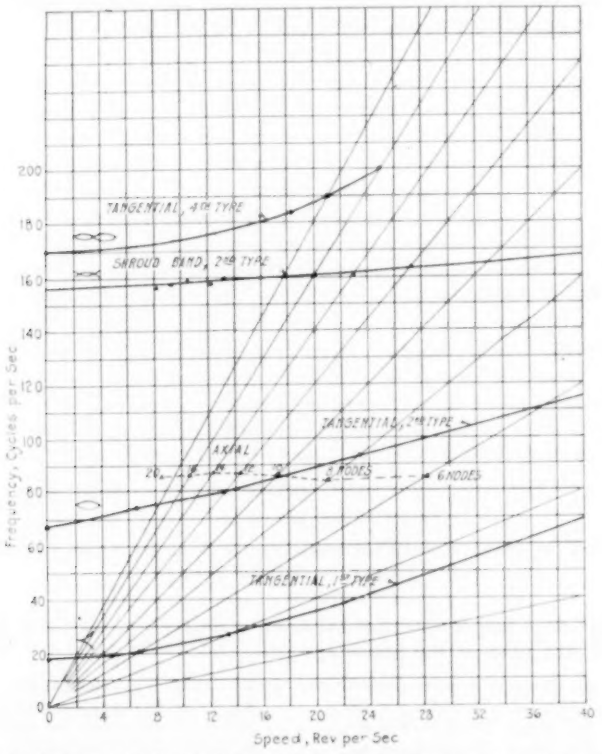

FIG. 9 FREQUENCY-SPEED DIAGRAM SHOWING TANGENTIAL AND AXIAL VIBRATION OF TURBINE WHEEL AND BUCKETS

the wheel, approximately half of the group under one shroud band vibrates in the opposite direction to the other half.

42 As has been mentioned in the paper, if the bucket wheel is rotated, the various frequencies of the bucket increase due to the stiffening action of centrifugal force.

43 Vibration of a bucket in one type or another occurs when the speed is such that the natural frequency of the bucket is an exact multiple of the revolutions per second.

44 A periodic force not truly sinusoidal will build up a bucket vibration of a frequency equal to any exact multiple of the frequency of the force. Therefore the presence of a pulsating force of this character with a frequency equal to the revolutions per second is sufficient to explain the occurrence of a vibration of any exact multiple of running speed.

45 The speed coefficient, B, in the equation $f_r = \sqrt{f_s^2 + BN_s^2}$ for the various types of bucket vibration is different for each type and in general higher than for axial nodal vibration.

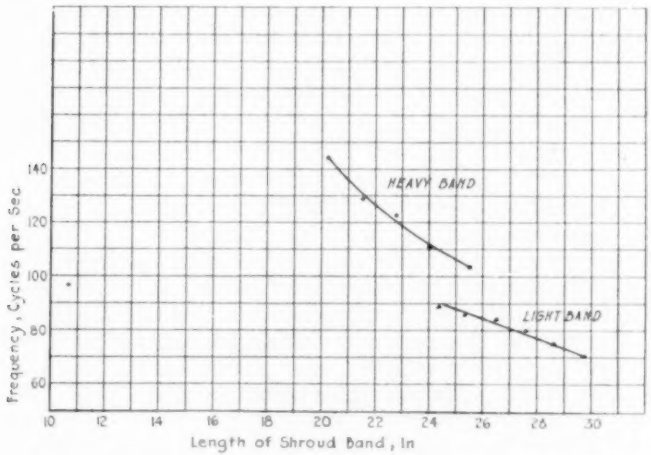

FIG. 10 EFFECT OF LENGTH OF SHROUD BAND ON SHROUD-BAND VIBRATION, SECOND TYPE

46 Standing frequencies taken on a reed 40 in. long and $\frac{3}{8}$ in. thick are given in the following table.

1st type,	9 cycles per sec.	5th type,	71 cycles per sec.
2nd type,	24 cycles per sec.	6th type,	123 cycles per sec.
3rd type,	30 cycles per sec.	7th type,	132 cycles per sec.
4th type,	63 cycles per sec.	8th type,	204 cycles per sec.

47 Since the various types of bucket vibration bear some proportional relationship to each other, it can be appreciated that only in the case of extremely long buckets must the higher types of bucket vibration be considered. In the other cases the greater fundamental frequency means proportionally greater frequencies for the higher types of bucket vibration so as to place them in the zone of safety. Our experience has been one of relatively few failures due to higher types of bucket vibration.

48 A frequency-speed diagram of representative results on a wheel with long buckets is shown in Fig. 9.

Shroud-Band Vibration

49 In addition to the various types of tangential bucket vibration, observations have been made of two types of bucket motion controlled by shroud-band vibration in an axial direction, that is, at right angles to the plane of the wheel and buckets.

50 In the first type a node is found in the center of the band section. The frequency of the band vibration is near that of the

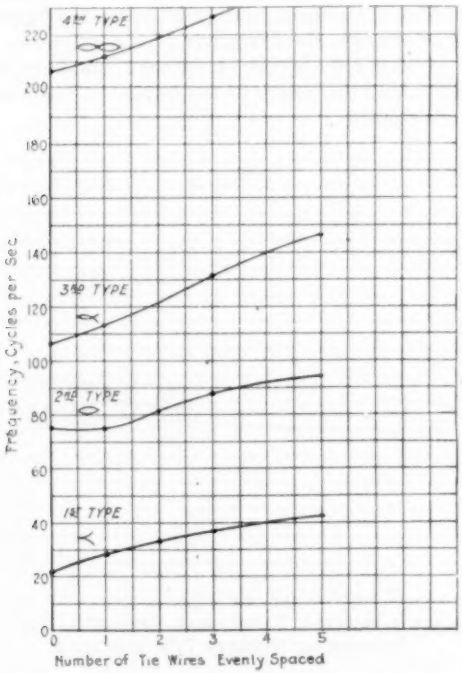

Fig. 11 Effect of Number of Tie Wires on the Various Types of Bucket Vibration

(Tie wires half the length of shroud-band sections so as to permit second and fourth types.)

bucket in the axial direction and is no doubt controlled by it. The band does not bend very much. This motion is a form of axial vibration having a number of nodes equal to the number of band sections.

51 The second type of shroud-band vibration occurs at a much higher frequency and consists of two nodes formed in the band section and is largely a shroud-band phenomenon. Only the outer ends of the buckets participate in the motion, while the band is bent quite considerably during vibration.

52 The frequency of the first type changes very slightly due to changes in length of cover, but the second type is affected quite appreciably, as shown in Fig. 10.

Tie Wires

53 Tie wires have long been used for lacing the buckets together. Their effect on the tangential frequency furnishes a means of tuning buckets, i.e., varying the tangential frequency.

54 The effect of tie wires on the odd-numbered types of bucket vibration is to increase the frequency of the vibration.

55 Fig. 11 shows the effect of different numbers of tie wires evenly spaced along the length of the bucket on the various types of bucket vibration.

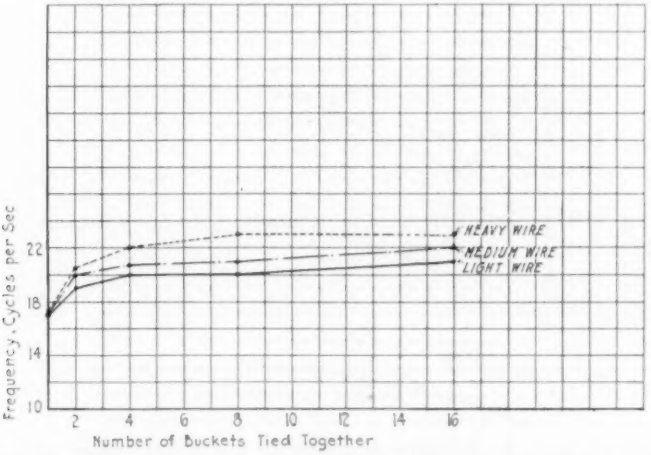

Fig. 12 Relation between Tangential Vibration, First Type, and Number of Buckets Tied Together

56 Fig. 12 shows the effect on the fundamental frequency of tying different numbers of buckets together and also different sizes of wire. As can be seen, tying four buckets together is about as effective as tying twenty or more buckets together.

57 The effect of a tie wire on shroud-band vibration is negligible, as would be expected. This is shown in Fig. 13.

58 On the other hand, a properly placed tie wire that ties together the same number of buckets as the shroud-band section is effective in eliminating the even-numbered types of vibration. But if each tie-wire length ties together half as many buckets as the shroud-band section, then the even-numbered types may exist due to pairing of the groups, and the tie-wire effect is only one of increasing frequency. This is shown in Fig. 11.

Relationship between Axial Nodal Vibration and Bucket Vibration with Long Buckets

59 Experience gained in wheel design has led to the formulation of certain rules which show that there is a connection between the so-called tangential bucket vibration and the axial nodal vibration discussed in the previous paper by Mr. Campbell on The Protection of Steam-Turbine Disk Wheels from Axial Vibration.

60 As will be recalled, it was shown that turbine wheels, i.e., disk and buckets, vibrate in certain nodal pictures having definite

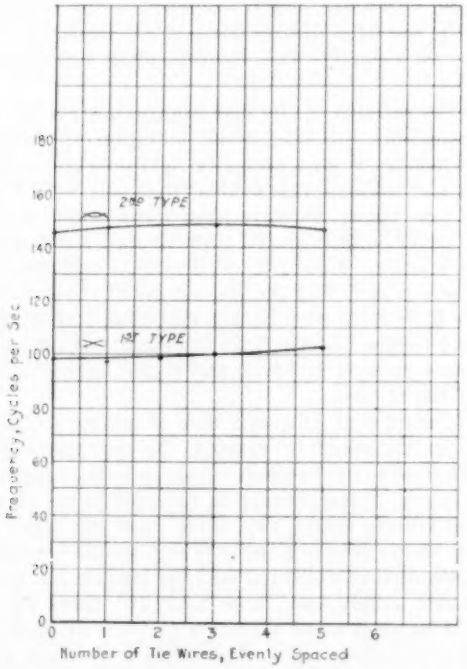

Fig. 13 Shroud-Band Vibration of the First and Second Types as Affected by Number of Tie Wires

natural frequencies. The nodal picture of each type formed in a stationary wheel was caused by two wave trains traveling in opposite directions having the same amplitude and speed. As the wheel was rotated, one wave train — the backward wave — slowed up relative to a fixed point in space while the other — the forward wave — speeded up. At a certain speed for each type of nodal vibration the corresponding backward wave was stationary in space. This speed was defined as a critical speed.

61 If a wheel with long buckets is vibrated axially at rest to determine the frequencies of four, six, eight, etc. nodes, it will be observed that at the higher numbers of nodes the standing frequencies become grouped closely together and approach a limiting value. This limiting value is the axial bucket frequency of the bucket where the bucket vibrates as a reed in the axial direction.

62 If the same set of buckets is applied to stiffer and stiffer disks, the frequencies of the lower nodal types which involve bending in the wheel increase quite appreciably, whereas the higher nodal types, which are largely confined to the buckets, increase only slightly, the series of frequencies approaching the axial bucket frequency as a limit. The corresponding critical speeds also increase.

63 As the stiffness of the disk approaches perfect rigidity, all of the frequencies for axial nodal vibration approach the axial bucket frequency. With a perfectly rigid dovetail connection to a disk of infinite stiffness, it is evident that only axial bucket vibration could take place. No wave phenomena could be present since no energy could be transmitted through such a stiff disk from one bucket to another. However, in all actual turbines, even if the rotor stiffness is such that the frequencies of the axial nodal types differ very little from pure bucket vibration, still the flexibility of the dovetail connection is such as to permit wave motion, which is further made possible by lacing and shroud connections along the lengths of the buckets.

64 If a wheel consisting of a disk of infinite stiffness but with ordinary buckets could be rotated, the resulting curve of axial bucket vibration would be similar in shape to the tangential bucket-vibration curve. Speeds would be recorded at which the buckets only would vibrate axially two, three, four, etc. times per revolution of the wheel. Now if a less stiff disk should be rotated, speeds would be recorded at which not only the buckets but also the wheel would vibrate axially two, three, four, etc. times per revolution. This would be axial nodal vibration in four, six, eight, etc. nodes composed of traveling waves. The respective critical speeds would be slightly lower than the corresponding points on the axial bucket-vibration curve.

65 The curve of axial bucket vibration is therefore the limit toward which the axial nodal-vibration curve approaches.

66 Heavy wheels with long buckets very nearly approach the above condition. The standing frequencies of the various axial nodal types are closely grouped below the axial bucket frequency. The critical speeds for each type of axial vibration result in a curve on the frequency-speed diagram that approaches very nearly to that of axial bucket vibration on the infinitely stiff disk. The disk effect is very small.

67 The design of wheels with long buckets resolves itself into determining the proper proportions of buckets so that the tangential and axial bucket frequencies give resulting critical-speed curves that provide ample margins of safety at running speed.

68 In the case of buckets of unsymmetrical section, the so-called "tangential" and axial vibrations are not exactly in the plane of the wheel and at right angles to it, but occur in perpendicular directions, neither of which is necessarily in the plane of the wheel. The result is two sets of nodal vibrations. In the lower set of four, six, eight, etc. radial nodes the vibration of the bucket is in a plane usually at a slight angle to the plane of the wheel. This is the type of vibration which Mr. Campbell defined as tangential bucket vibration. In the upper set of four, six, eight, etc. radial nodes the vibration is almost at right angles to the plane of the wheel, and this type was described in Mr. Campbell's earlier paper and called axial vibration. Both of these sets of vibrations are due to traveling waves in the wheel and buckets, and the resulting critical speeds are due to the backward wave being stationary in space.

69 As the section of the buckets progressively changes from the unsymmetrical to the symmetrical type, the characteristics of tangential vibration involving different radial nodes approach the characteristics of pure bucket vibration in the plane of the wheel. During this change the influence of the disk on the vibration becomes less and less, and finally disappears.

70 If the buckets are symmetrical with respect to the plane of the wheel, then only bucket vibration exists and the resulting curve is bucket vibration. At certain speeds the bucket vibrates tangentially two, three, four, etc. times per revolution, as Mr. Campbell's notes have well illustrated.

71 If the buckets are slightly unsymmetrical, nodal vibration, mostly tangential but with a small axial component, takes place in which traveling waves are present. The resulting critical speeds are due to the backward-traveling wave in the wheel being stationary in space.

72 The curve of true tangential bucket vibration is then the limit of nodal vibration during which the buckets vibrate mostly in the tangential direction.

73 Thus the two curves of pure bucket vibration with an infinitely stiff disk, one for the axial and one for the tangential type, are actually the limiting curves for axial and tangential vibration of the disk and buckets in radial nodes, the deviation from these curves depending only on the degree of flexibility of the disk.

74 In fact, no set of buckets in a wheel is absolutely symmetrical. The tangential bucket vibration can then be considered as nodal vibration in which the disk participates very slightly. The result is that the standing frequencies of the various nodal types

have practically the same value. The resulting curve of critical speeds is one for nodal vibration with a very stiff disk.

75 It is interesting to note that as the bucket vibrates tangentially at a critical speed it repeats the same path each revolution. Any part of it describes a circumferential wave which is stationary in space, so that again we have the condition of a wave stationary in space, although this is a longitudinal wave while the axial vibration consists of transverse waves.

Short Buckets

76 As the length of the bucket decreases, the effect of the disk on axial nodal vibration becomes more pronounced and a different phenomenon takes place.

77 The critical-speed curve of axial nodal vibration on the frequency-speed diagram for a plain disk or a disk with rim, at the lower nodes, is one in which the critical speeds increase with the number of nodes or waves as shown in Curve I, Fig. 14. This is opposite to the case where the bucket effect predominates and causes the critical speeds to decrease when the number of nodes increases as shown in Curve IV, Fig. 14.

78 Actual turbine wheels therefore show a combination of the above phenomena. Wheels with short buckets give resulting axial nodal pictures in which the critical speeds of the lower numbers of nodes such as four, six, eight, etc. increase with the number of nodes according to the rule for a plain disk like Curve I, Fig. 14, the bucket effect being very small on these nodes. However, in the same wheel the critical speeds for such types as thirty nodes and higher, which are mainly axial bucket phenomena, decrease with the increasing number of nodes according to the rule for buckets. The frequencies of the higher nodes finally approach the bucket axial frequency as a limit.

79 Between the two extremes there is a condition extending over a number of nodes during which a transition takes place from the plain-wheel characteristics to the plain-bucket characteristics.

80 For any particular disk, the higher the axial bucket frequency the higher the number of nodes at which this transition takes place.

81 Fig. 14 is a frequency-speed diagram on which the points represent critical speeds for the various types of nodal vibration.

82 Curve I is a curve of a plain disk. As can be seen, the critical speeds progressively increase as the number of nodes increases.

83 Curve IV gives data from a wheel having a bucket 34 in long. This is almost a true axial bucket vibration curve, and, as can be seen, the critical speeds decrease with the increasing number of nodes.

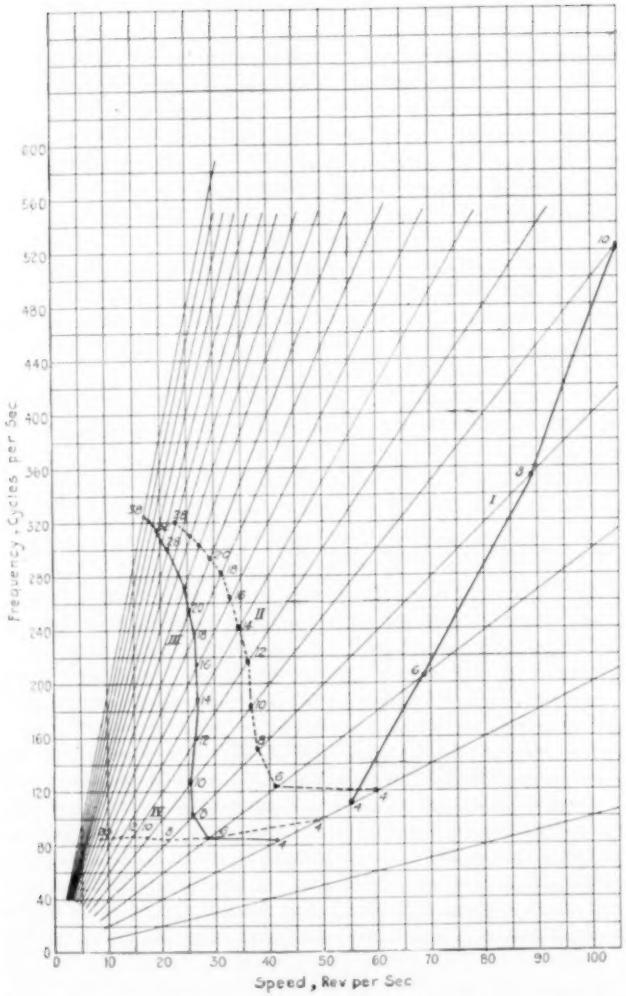

FIG. 14 FREQUENCY-SPEED DIAGRAM SHOWING CRITICAL-SPEED CURVES CHARACTERISTIC OF WIDELY DIVERGENT TYPES OF TURBINE WHEELS

(Numbers on curves indicate number of radial nodes.)
I, Plain disk without buckets.
II, Disk with medium-length buckets.
III, Thinner disk with same buckets as II.
IV, Thick disk with long buckets.

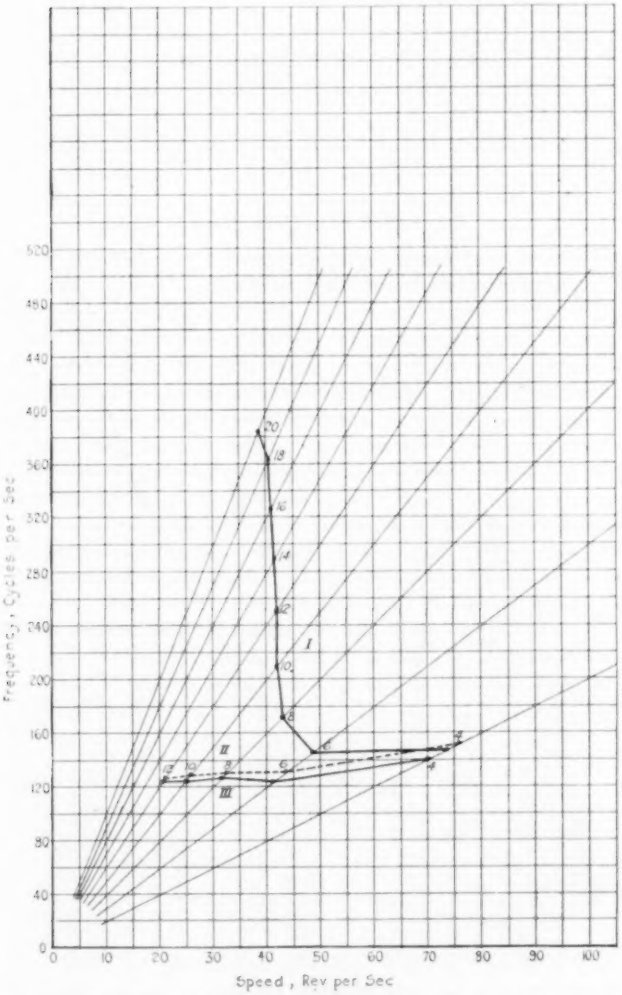

FIG. 15 FREQUENCY-SPEED DIAGRAM SHOWING CRITICAL-SPEED CURVES CHARACTERISTIC OF THREE DIFFERENT WHEELS

(Numbers on curves indicate number of radial nodes.)
I, Small thin disk with medium-sized buckets.
II, Thick disk with long buckets.
III, Thinner disk with same buckets as II.

84 Curve III is one of a wheel having an 8-in. bucket and a pitch diameter of 84 in. The web thickness was $\frac{5}{8}$ in. As can be noted, the disk effect is noticeable up to the fourteen nodes and then the bucket effect manifests itself and the critical-speed curve becomes like that of bucket vibration, approaching the axial bucket frequency as a limit.

85 Curve II is a wheel having a bucket similar to that shown in Curve III, but a heavier turbine disk. As noted, the heavier disk raises the critical speeds of the lower nodes, but has very little effect on the higher nodes as this vibration is practically all in the buckets.

86 Fig. 15, Curve I is the result of a test on a wheel consisting of a medium-length bucket on a small thin disk. The combination effect is one of having the critical speeds of a considerable number of nodes within a few per cent of each other.

87 Fig. 15, Curves II and III are the results of tests on two wheels having long buckets. The resulting curves are very near that of true axial bucket vibration. The wheel represented by Curve III had a web thickness of $2\frac{1}{4}$ in., while that of Curve II had a web thickness of $3\frac{1}{4}$ in. It will be noticed that the increasing of the disk stiffness has very little effect on the wheel critical speeds for axial nodal vibrations.

88 In Mr. Campbell's previous paper the methods used for the design and testing of wheels to protect them against axial vibration were fully described. Since then there has been a real advance in the art of designing large wheels for the exhaust stages of steam turbines, largely made possible by these further investigations. The same careful consideration is now also given the turbine buckets to protect them against tangential bucket vibration.

89 The importance of the pure component vibration frequencies of the bucket as the limiting values controlling the tangential frequencies on the one hand and the axial frequencies on the other hand cannot be overestimated. This relationship is most useful because pure bucket vibrations are more easily susceptible to exact calculation than vibrations of the wheel as a whole, and preliminary tests on a few buckets are quickly made previous to the final design of a wheel. All of this knowledge enables both the tangential- and axial-vibration characteristics of a wheel design to be predicted in advance with more certainty than ever before.

90 On the other hand, it cannot be overemphasized that the complete running tests as established by Mr. Campbell for the final check on the critical-speed margins of wheels are absolutely essential. The methods of tuning described in connection with the paper on axial vibration are regularly resorted to, but better understanding of the principles has resulted in confidence in exhaust-stage wheels of a size never before built.

DISCUSSION

BENJAMIN FOX.[1] The authors' latest contribution to our knowledge of the vibration of turbine buckets is very important. It has been found that as the size of the wheel and the length of the bucket are increased, difficulties due to tangential bucket vibrations appear long before there is any trouble from axial vibrations of buckets or nodal vibrations of disks.

The paper sets forth very clearly the general results of the research; further details on certain phases would be interesting and useful.

In Par. 5 a formula is given for the frequency of a bar in terms of stiffness and mass; and in Pars. 18 and 20 is given a method for estimating the frequency of a bucket from the known frequency of a geometrically similar bucket.

The fundamental formula of Par. 5 can be applied directly only to buckets that are of uniform section throughout their length. The frequency of buckets with taper backs or otherwise varied sections cannot even be approximated by this formula. The method of estimating by geometrical similitude is useful, but it is not applicable to buckets of new proportions.

A method for calculating the standing frequency of a turbine bucket of any shape has been developed and used in the engineering department of the Bethlehem Shipbuilding Corporation. It is based on the well-known relation that the kinetic energy of a vibrating bar in its mean position is equal to the potential energy of the bar in its extreme position. Lord Rayleigh has shown that, for the purpose of calculating frequency, the deflection curve of the vibrating bar in its extreme position may be assumed to be the same as the curve of static deflection of the bar under its own weight. This leads to the general formula:

$$f = \frac{g^{\frac{1}{2}}}{2\pi} \left(\frac{\int wy dx}{\int wy^2 dx} \right)^{\frac{1}{2}} \quad \ldots\ldots\ldots [4]$$

where f = frequency of the bar (one node)
g = acceleration due to gravity
w = weight per unit length of bar at any point
y = static deflection of bar at any point.

For the usual turbine bucket of varying section the value of y in terms of x cannot be expressed mathematically. One method of using the foregoing relation in such cases is to evaluate the integrals arithmetically. The amount of numerical work involved is rather large; with suitable forms an experienced computer gets the result for a new design in about four hours. The results obtained have been checked in many cases by actual measurement

[1] Engineering Department, Fore River Plant, Bethlehem Shipbuilding Corpn., Ltd., Quincy, Mass. Mem. A.S.M.E.

of the frequency and have been found correct within the limits of error of the available measuring equipment. Incidentally, they have revealed errors in methods of approximation formerly used.

The phenomenon of resonance between the augmented frequency of a turbine bucket and the speed of rotation of the wheel has been suspected for some time in connection with certain bucket failures. In some cases resonance multiples much higher than 3 are indicated, as when there are several interruptions of the steam flow during one revolution of the wheel. For instance, when there are two bridges in the nozzle diaphragm a bucket will get two impulses each revolution, and, because of this, there will be resonance when the ratio of augmented frequency to speed is 2, 4, 6, etc.

The influence of the rotation of the wheel upon the bucket frequency was investigated analytically and formulas were derived for two cases for the value of B in the equation

$$f_r^2 = f_s^2 + BN_s^2$$

Let R = radius of rotation of the bucket, from center of wheel to fixed end

l = length of vibrating portion of bucket (for vibration with one node).

Then, for a bar of uniform section throughout its length,

$$B = \frac{81}{52}\frac{R}{l} + \frac{61}{52} \quad \ldots\ldots\ldots [5]$$

and for a rectangular bar of constant width, tapering uniformly in thickness to nothing at the free end,

$$B = \frac{2R}{l} + \frac{4}{3} \quad \ldots\ldots\ldots [6]$$

The derivation of the formulas is too lengthy to be included in this discussion. The values of B given above probably define the limits of variation for the usual buckets of varying section without banding. The influence of the banding undoubtedly increases the coefficient. It would be interesting to know whether the experimental results obtained by the author confirm the values derived analytically.

In the case of turbine buckets of varying section a rational general formula is unattainable, but the method of arithmetical integration mentioned above has been extended so that it is possible to evaluate this coefficient and so estimate, in advance of tests, the frequency of a bucket at various speeds of the wheel, and consequently also the resonant speeds.

These points are presented with the thought that perhaps the author may be able to disclose a less laborious method of calculating both the standing and the running frequencies of buckets of varying section, where the vibration characteristics for geometrically similar buckets are not available.

S. H. WEAVER.[1] Mr. Heckman very clearly explains the influence of the frequency of the bucket vibration upon the frequency of the wheel. This influence of the bucket frequency has not been fully appreciated, particularly in connection with the problem of estimating the frequency of the wheel.

The writer was one of those who attempted to predetermine the wheel frequency from the design. Professor Stodola's theory given in *Schweizerische Bauzeitung*, May, 1914, was used, and his formula charted for rapid application. The result was that the estimate on several wheels agreed closely with tests, but was followed by other wheels which gave an absurd difference between calculation and test. Hence no confidence could be placed in the estimated values.

A number of persons then became engaged in a critical study of Stodola's work, and the only thing found was that the potential energy of the buckets was not included in the vibration formula. For small- or medium-length buckets the energy was negligible, but for longer buckets the energy quickly became of important value. We were unable properly to insert the potential energy of the bucket in Stodola's formula.

Tests were made on full-sized forged steel disks of true hyperbolic shape, with various sizes of rim from extremely large to no rim, and in all cases without buckets or dovetail, the ideal condition of theory. It was found that Stodola's vibration formula agreed with tests for the disk without rim or buckets; also for a disk with a rim of practicable proportions for one row of buckets; but when the rim was large enough for two rows of buckets, an error was found. The tests gave the second indication that the error in the estimate of the frequency of a complete wheel was to be found in the consideration of the buckets.

By this time we had collected a quantity of test results on actual wheels which were elaborately correlated and indexed so that the frequency of a wheel of new design could be closely estimated by comparison with tests; and as the necessity for a complete formula for estimating the frequency no longer existed, our studies were discontinued.

Mr. Heckman explains, and the test results in Figs. 14 and 15 show, the influence of the bucket frequency upon the wheel frequency. When one considers the wheel as a combination of two elastic members, the disk with a frequency of its own, supporting the second member which consists of the buckets with a different natural frequency, the combination of these two unlike members in the wheel presents a somewhat complex problem to calculate.

Equation [4] as given by Mr. Fox is not an approximation but a real vibration equation. We have developed it in the same

[1] Turbine Engineering Department, General Electric Co., Schenectady, N. Y.

manner as outlined by Mr. Fox and in using it for the last 18 months, we have found it very reliable. The approximate part is the equation for y which must be assumed, as the theoretical value for a bucket with constant section will not hold for the tapered bucket. The curve we use is of the exponential type and the exponent has been derived from tests. Considerable experimental work was necessary to obtain the curve, but since it has been obtained we have had confidence in it.

G. L. KNIGHT.[1] The rapid growth of the demand for electric power has brought about the development of turbines of sizes heretofore not considered practical or commercial. Paralleling this growth is a similar growing demand for reliability. These demands are not mutually consistent where design is empirical, certainly not when such great advances have been made as have been witnessed in the last few years. It is such work as discussed by the authors of this paper that enables the industry to make the progress it has and gives utility engineers and executives confidence in the use of large units. It is natural as machine sizes are increased without experiencing major difficulties, to forget that behind all this is the pioneer investigation and research which places that design on firm ground. The real credit for success should go to such men as the authors. The task for those in the operating field is simplified, in fact, only made possible, through their painstaking labors.

S. A. MOSS.[2] So far as the writer knows, the first difficulty with tangential vibration of buckets occurred at the Lynn works of the General Electric Company some years ago. The nature of the phenomenon was found, but we did not have at that time the formulas which Messrs. Campbell and Heckman have developed. At that time Mr. Richard H. Rice, turbine engineer, made a rule that with tall buckets the nozzle arrangements be absolutely symmetrical; there must be no bridges and no chance for partial admission of steam to affect the lower end of the turbine. It may be pertinent to inquire, if where there is total admission, whether such a rule, carried to the limit, would eliminate the tangential vibration, even at a resonant speed.

A. DEVAUD.[3] In answer to Mr. Fox's question about the calculation of the natural frequency of vibration of turbine blades, it may be of interest to state that the use of a single deflection curve,

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[2] Engineer, Mechanical Research Dept., River Works, General Electric Co., West Lynn, Mass. Mem. A.S.M.E.

[3] Turbine Engineering Department, General Electric Co., Schenectady, N. Y.

while satisfactory for approximate results, cannot be applied rigorously to determine the frequency of tapered buckets of widely different types. It should be remembered that, mathematically speaking, the actual shape of the deflection curve is a function y of x that will satisfy the differential equation:

$$\gamma\omega^2 Ay = E\frac{d^2}{dx^2}\left(I\frac{d^2y}{dx^2}\right)$$

where

γ = mass per unit volume of material
E = modulus of elasticity of material
ω = frequency (in radians per unit of time), a constant with respect to x
A = cross-sectional area at abscissa x
I = moment of inertia of cross-sectional area at abscissa x
y = ordinate of deflection curve at abscissa x.

This equation disregards the slight shape deviation caused by the fact that, in general, the principal axes of inertia are not necessarily parallel in the different cross-sections throughout the blade.

A graphical method of calculation, based on Stodola, has been developed by the writer to determine these actual deflection curves, not only for vibration of the fundamental type, called "first" type in Mr. Heckman's paper, but also for the second, third, fourth, and fifth types.

This method is based on the analogy existing between natural frequency of vibration and critical speed, and is derived from the fact that at its critical speed any balanced body is in a state of neutral equilibrium for any deflection within the limit of elasticity. The beauty of the method is that the shape is found without having even to mention the complicated differential equation referred to above.

W. C. HECKMAN.[1] Throughout the investigations described in this paper there has been a definite field for theoretical work and another for experimental work in the problem of bucket vibration. Theoretical investigations were carried satisfactorily to a certain point beyond which experimental information became necessary. The combination of the results of the two investigations gave the answer to the problem.

In answer to Mr. Fox's question concerning the possibility of calculating the value of the speed coefficient B in the equation

$$f_r = \sqrt{f_s^2 + BN_s^2}$$

it is practically impossible to calculate the B value with any degree of accuracy. The formula given by Mr. Fox considers

[1] Closure to discussion.

only the stiffening effect of centrifugal force on the vane section. It disregards the effect of the bucket dovetail fit which may cause a considerable variation in the value of B.

In fact, actual tests under controlled conditions have shown the possibility of assembling a certain design of bucket in a particular wheel so as to obtain a 100 per cent variation of B values. Whether the B value was high, low, or actually negative depended upon the degree of tightness of dovetail fit. In the latter case the running frequency was actually lower than the standing frequency.

While such variations are extreme, we cite them simply to call attention to the necessity of test control of the product in every case, reliance on calculation alone being inadequate.

Regarding the problem of excitation of tall buckets sensitive to resonant vibration raised by Dr. Moss, it is agreed that if there were positively no exciting force present, we should have no vibration because vibration depends on an exciting force for the energy supply. However, if we were to remove all known exciting forces in the turbine we should not be sure that we had removed such possible causes as the periodic variation of torque in the generator when the phases are slightly unbalanced.

We like to consider the problem of vibration as including, first, the tendency of the bucket to vibrate, and, second, the existence of proper exciting forces. The only satisfactory remedy is to guard against the tendency of any bucket to vibrate at or near running speed because we never can be sure that all exciting forces have been removed.

In the first part of the paper Mr. Campbell limits bucket failures to 2, 3, and 4 cycles per revolution. This is based entirely on the experience available at that time. We have had an occasional failure due to a higher frequency of vibration, but we have always been able to associate such failures with an exciting force of some extraordinary nature.

By the application of the principles and standards set forth in the paper the number of bucket failures due to tangential bucket vibration has been rendered insignificant when one considers the large number of blades in successful operation. A continued policy of careful design and actual testing of buckets is demanded by the seriousness of turbine outages due to bucket failures.

No. 1976

VIBRATION PHENOMENA OF A LOADED UNBALANCED SHAFT WHILE PASSING THROUGH ITS CRITICAL SPEED

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and

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Junior Member of the Society

The purpose of this paper is to present in a clear manner with experimental demonstration a discussion of the peculiar vibration phenomena to which an unbalanced, loaded shaft is subject while passing through its critical speed. First, the basic theory is given from a somewhat new viewpoint,[2] after which an experimental demonstration, which gives a quantitative check on the theory, is described.

MANY problems in shaft whirling may be handled by a method analogous to that used in the solution of alternating-current circuit problems, in the sense that transient oscillations are disregarded and only the steady condition of exactly repeating vibrations is considered. This type of analysis has proved very useful in the case of alternating-current problems, and has similar advantages for the solution of certain shaft-vibration problems.[3] It consists of solving directly for the condition of steady vibration by means of the diagram of forces, which can readily be done by making use of the magnitudes, directions, and phase angles of the forces involved.

[1] Research Laboratory, General Electric Co.

[2] Since the writing of this paper, it has been called to the authors' attention that an essentially similar method is discussed by A. Stodola in his Dampf und Gas Turbinen, p. 362.

[3] This method of handling shaft-vibration problems has been used independently by H. D. Taylor of the General Electric Co. in work not yet published. Furthermore, results of experimental work by Mr. Taylor and Dr. B. L. Newkirk on this same problem of shaft behavior at critical speeds, except that a horizontal high-speed model was used, proved very helpful in the writing of this paper. The authors wish to call particular attention to this work, as it was evidently planned as well as carried through before theirs.

Presented at the Annual Meeting, New York, November 30 to December 4, 1925, of The American Society of Mechanical Engineers.

2 The particular problem of shaft whirling to be considered in this article is reduced to its simplest terms by assuming the rotor to be a disk-shaped flywheel, carried by a straight, weightless shaft. In a perfectly balanced rotor of this type the center of gravity of the flywheel must coincide with the central axis of the shaft which carries it.

3 Unbalance may be produced, for example, by drilling a hole in one side of the disk. The center of gravity G of the disk is then displaced a small distance e from the center line S of the shaft.

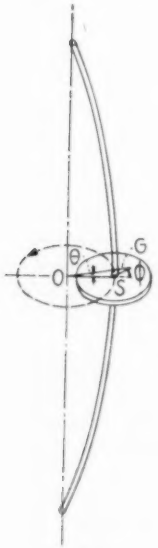

FIG. 1

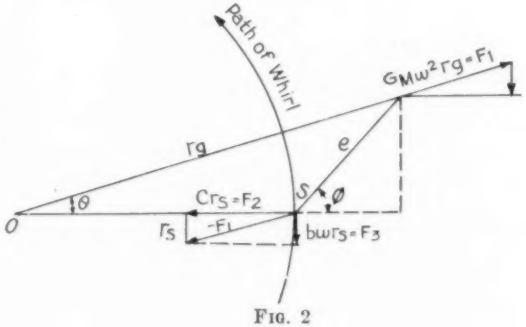

FIG. 2

FIG. 1 SHAFT WHIRLING DUE TO UNBALANCE, SHOWING RELATIVE POSITIONS OF CENTER OF MASS OF FLYWHEEL G, CENTER OF SHAFT S, AND CENTER LINE OF BEARINGS O, FOR CASE OF ROTATION BELOW CRITICAL SPEED

FIG. 2 EQUILIBRIUM DIAGRAM FOR CASE OF FIG. 1

Consequently when the shaft commences to rotate, G begins to describe a circle of radius e about S, which develops a centrifugal force whose magnitude would be $M\omega^2 e$ with a perfectly stiff shaft (M = mass of flywheel). However, since the shaft itself springs slightly, the centrifugal force becomes greater than this. In other words, S is displaced outward from the center of rotation so that both S and G describe circles about the center of rotation O, the static position of S, or the point of intersection of the line drawn through the bearing centers perpendicular to the plane of motion of S and G. The relations of these three points are shown in Fig. 1

and also in Fig. 2, as they would appear on a surface passed through the plane of the flywheel.

4 For the steady state of vibration at a given speed of rotation, the points O, S, and G must be definitely related to each other, and the angle must have a particular value. Since the forces acting at G and S are known in direction and magnitude, as shown in the figure, a direct solution for amplitude of whirl of the shaft r_s and the phase angle ϕ can be obtained, expressed as a function of the speed of revolution. For purposes of analysis (see Fig. 2), let

$$\left.\begin{aligned} F_1 &= M\omega^2 r_g \\ F_2 &= cr_s \\ F_3 &= b\omega r_s \end{aligned}\right\} \quad \ldots\ldots\ldots\ldots [1]$$

where F_1 is the centrifugal force of the mass M as it whirls about O and is considered as though acting at its center of gravity G. F_2 is the elastic restoring force exerted by the deflected shaft tending to bring its center S back to its undeflected position O. F_3 is the equivalent of a frictional force resisting the whirling motion of the shaft. It is therefore tangent to the path of whirl and vanishes when the deflection of the shaft r_s vanishes. For purposes of analysis it can be represented by F_3 applied at S since such a force develops a counter torque resisting the whirl of the shaft and flywheel as a whole about O, and does not tend to resist the rotation of the shaft independently. S is the geometrical center of the whirling system, where such an equivalent force should naturally be applied. In this problem it will be assumed to have a value which depends upon the velocity of motion of S about O equal to $b\omega r_s$, where b is the damping constant.

5 When the shaft is whirling at a constant speed, complete equilibrium must exist both for translation and for rotation.

6 For translational equilibrium, the force F_1 must exactly balance the resultant of F_2 and F_3 as indicated in the figure. For rotational equilibrium the moment of the resultant of these three forces about O must balance the driving couple, or, since F_1 and F_2 both pass through O, the moment of F_3 about O must balance the driving couple.

7 From Fig. 2, by similar triangles,

$$\frac{e \sin \phi}{r_g} = \frac{F_3}{F_1} \quad \text{and} \quad \frac{e \cos \phi + r_s}{r_g} = \frac{F_2}{F_1}$$

from which

$$\sin \phi = \frac{F_3 r_g}{F_1 e} \quad \text{and} \quad \cos \phi = \frac{F_2 r_g}{F_1 e} - \frac{r_s}{e} \quad \ldots\ldots [2]$$

$$\tan \phi = \frac{\sin \phi}{\cos \phi} = \frac{\dfrac{F_3 r_g}{F_1 e}}{\dfrac{F_2 r_g}{F_1 e} - \dfrac{r_s}{e}} = \frac{F_3 r_g}{F_2 r_g - F_1 r_s} \quad \ldots [3]$$

Substituting from [1]

$$\tan \phi = \frac{b\omega}{c - M\omega^2} \quad \ldots\ldots\ldots [4]$$

From the value of $\tan \phi$

$$\sin \phi = \frac{b\omega}{\sqrt{(c - M\omega^2)^2 + b^2\omega^2}} \quad \ldots\ldots [5]$$

From [2]

$$\sin \phi = \frac{F_3 r_g}{F_1 e} = \frac{b r_s}{M\omega e} \quad \ldots\ldots\ldots [6]$$

Combining [5] and [6] and solving,

$$r_s = \frac{Me\omega^2}{\sqrt{(c - M\omega^2)^2 + b^2\omega^2}} \quad \ldots\ldots [7]$$

Equations [4] and [7] are the well-known expressions[1] for the phase angle ϕ and the shaft deflection r_s expressed as functions of the velocity of rotation ω. In the neighborhood of $\omega = \sqrt{\frac{c}{M}}$ both ϕ and r_s pass through rapid changes (see Figs. 10 to 13), and when

$$\omega = \omega_c = \sqrt{\frac{c}{M}} \quad \ldots\ldots\ldots [8]$$

$\phi = 90$ deg. and r_s reaches almost its peak value r_{sc}

$$r_{sc} = \frac{Me\omega_c}{b} = \frac{e}{b}\sqrt{Mc} \quad \ldots\ldots [9]$$

This condition is known as the shaft critical speed, being exactly analogous to the critical voltage value at electrical resonance.

8 In practice, the friction constant b is often sufficiently large so that the peak value of r_s does not correspond to a dangerous whirl.

9 A remarkable change, however, takes place when the critical speed has been exceeded. An almost complete reversal of the positions of G and S occurs. This is shown in Figs. 3 and 4. The phase angle ϕ becomes greater than 90 deg. with the result that the center of mass G of the flywheel swings inward toward O and may take a position almost between O and the shaft center S. A consideration of the force diagram (Fig. 4) shows how the equilibrium is possible.

10 One way of looking at the tendency of the center of mass M to move in toward the axis of rotation is to recall that a free rotating mass always revolves about its center of gravity unless

[1] See Föppl, *Vorlesung über Technische Mechanik*, vol. 4, p. 281, and H. H. Jeffcott, Lateral Vibrations of Loaded Shafts in the Neighborhood of a Whirling Speed. *Phil. Mag.*, March 1919, p. 304.

restrained from doing so. When the speed of the unbalanced flywheel becomes high enough, this tendency overcomes all other forces, and the center of gravity moves in close to the center of rotation, forcing the elastic shaft to move around it. The action is similar to that of a button which is made to spin to and fro upon the center of a span of string by pulling the string alternately so as to twist it first one way and then the other. The button will always rotate about its center of mass whether the cord passes through it or not.

11 It is thus seen that such a problem may be directly solved by the consideration of equilibrium of the forces involved without

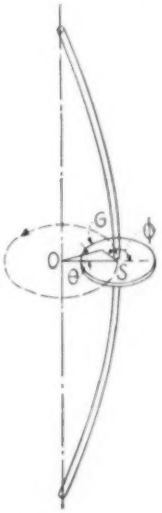

Fig. 3

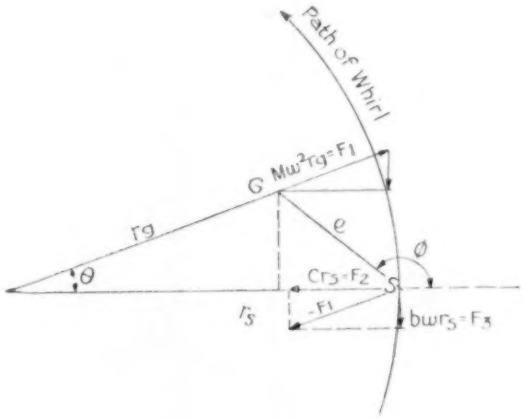

Fig. 4

Fig. 3 Same as Fig. 1 but for Rotation above Critical Speed, Showing Change in Position of Center of Gravity of Flywheel

Fig. 4 Equilibrium Diagram for Case of Fig. 3

undertaking the solution of differential equations. The method is quite powerful, and may be used to handle other problems of greater difficulty than this one.

Apparatus and General Method

12 In order to check quantitatively this theory over a considerable range of speeds for various degrees of damping and of unbalance, a model was made containing the basic elements of an actual rotating machine but so designed that such factors as mass, degree of unbalance, and degree of damping could be varied at will.

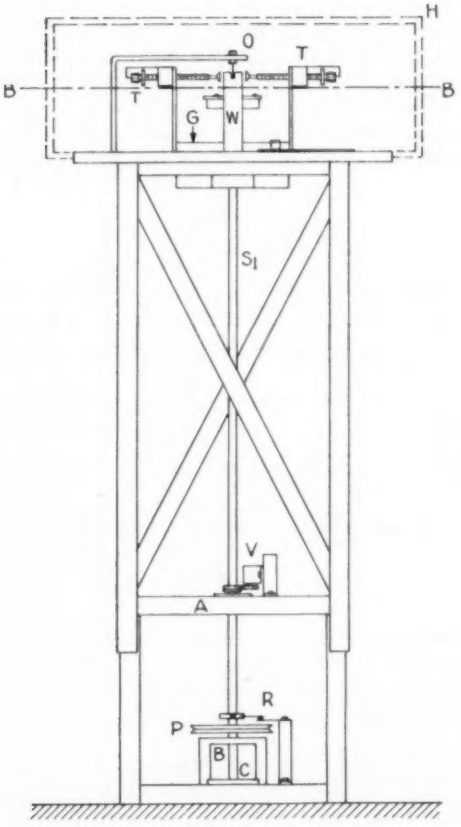

Fig. 5a

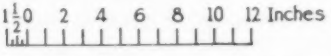

FIG. 5 DETAILED DRAWING OF APPARATUS

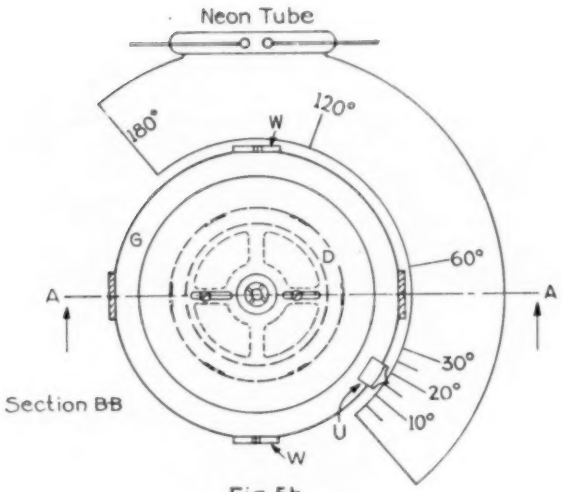

Fig. 5b

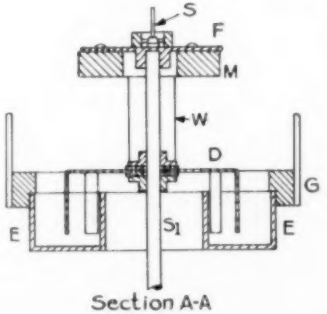

Fig. 5c

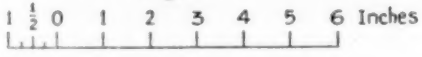

FIG. 5 (Continued)

13 In order to facilitate accurate observation, the model, drawings, and photographs of which are shown in Figs. 5 to 8, was so designed that the critical speed was low and the amplitude of deflection was large. For the same reason the flywheel M was carried on the end of an overhanging shaft instead of between the bearings. The diameter of the flywheel was so small compared with the shaft overhang that the effect of the gyroscopic action present in such an arrangement was negligible.

14 The steel shaft S, $\frac{5}{16}$ in. by 36 in., running in ball bearings at A and B and on the smooth step bearing C, Fig. 5*a*, carried at its top a disk-shaped flange F. A variable-speed motor, run from a storage battery to insure constant voltage, drove this shaft through an intermediate shaft and the pulley P.

15 The flywheel M was attached to the flange F by screws in slots by means of which the flywheel might be given a slight

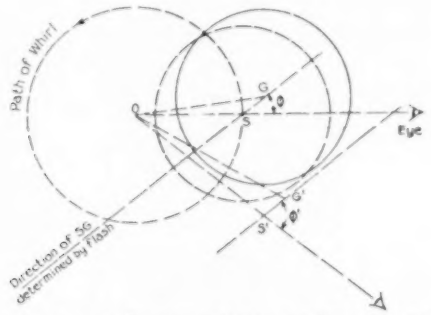

FIG. 6 DIAGRAMMATIC REPRESENTATION OF METHOD OF MEASURING PHASE ANGLE

eccentricity corresponding to a desired degree of unbalance. Flywheels of several different weights were also provided giving different critical speeds. The eight paddles of the damper D, Fig. 5*c*, through the central hole of which the shaft revolved freely, dipped into the annular oil canal E, containing a mixture of manila gum and castor oil.

16 The method of obtaining the required measurements is illustrated by the diagram of Fig. 6. The axis S of the shaft is marked by the vertical pointer which it carries, as shown in Fig. 5*c*. See also Fig. 1. The point of zero shaft deflection O about which the shaft whirls is marked by a downward-projecting pointer O, (Fig. 5*a*) attached to a fixed arm which reaches over from the side. When there is no deflection of the shaft the two pointers should line up, one pointing up and the other down. When a whirl builds up, as shown greatly exaggerated in the diagram of Fig. 6, the line joining the shaft center S and the center of gravity G of the flywheel M makes an angle ϕ with the line passing through

the downward pointer O and the upward pointer S. In order to observe this angle during the whirling motion, the upper part of the apparatus was hooded to screen off outside light, and was instantaneously illuminated once every revolution by a neon glow

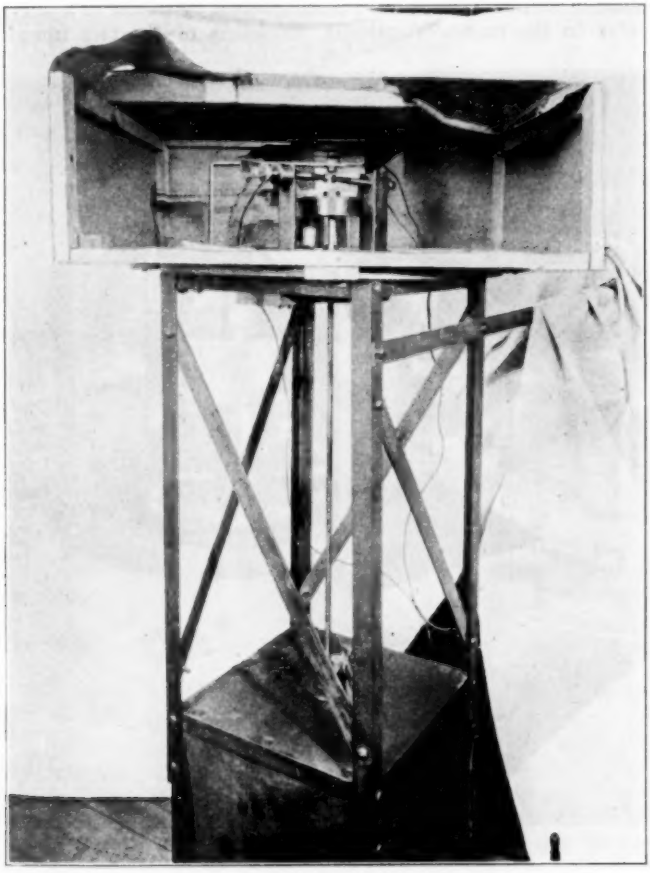

FIG. 7 PHOTOGRAPH OF APPARATUS

lamp, shown in Fig. 5*b* and Fig. 8. The illumination was produced by an electric contact R, Fig. 5*a*, at the base of the shaft. Since this contact necessarily always took place at the same angular position of the shaft, the illumination was produced only when the line of eccentricity $e = SG$ was in a certain direction, irrespective of the direction of deflection of the shaft $r_s = OS$. For instance

in Fig. 6, when the speed of revolution is raised nearer to the critical speed, SG moves over to $S'G'$, but still points in the same direction.

17 The direction of shaft deflection OS is located readily with the eye by finding where the pointers O and S line up at the instant of illumination. This is accomplished by means of a device similar to the mariners' pelorus, consisting of the two upright

FIG. 8 DETAIL OF TOP OF APPARATUS

sighting vanes W, Fig. 5, attached to the base G, capable of rotating about the center line of the bearings on which the indicator U is mounted by means of which the angle can be read from a scale attached to the frame of the machine. When the variations of the angle ϕ are thus determined, the amount of the deflection OS, or the amplitude of the whirl, is measured by the pair of micrometers T shown in Fig. 5*a* and also in Figs. 7 and 8. The speed is measured by the speed counter V.

18 By means of this apparatus, a quantitative check was obtained on the basic theory presented, and this was done for several degrees of damping. The results are shown in the curves of Figs. 10 to 13. The smooth curves are plotted from Equations [4] and [7] and the circles and crosses indicate actual observations made.

Experimental Procedure

19 The accuracy of the results obtained depended upon the exactness of certain preliminary adjustments. As it was impossible to obtain a perfectly straight shaft of the dimensions required, two centering devices were necessary — one at the damping-paddle bearing and another holding the vertical pin on the end of the shaft. They were so adjusted that the damping paddle and the pin

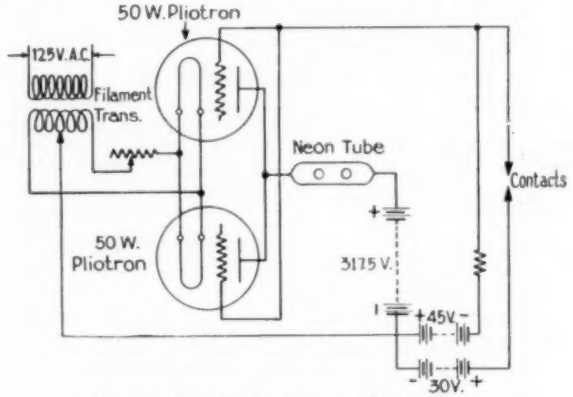

Fig. 9 Circuit Diagram of Neon Tube

S both remained accurately centered during such slow rotation that deflections due to centrifugal force were negligible.

20 The damping fluid used, a mixture of manila gum and castor oil, was chosen because its variation of viscosity with temperature was small, and also because its stiffness could be changed at will by varying the proportions of the mixture. Heavy machine oils gave erratic results because of their great variation in viscosity with temperature. Before making a run, the eccentricity of the flywheel was so adjusted that the maximum deflection, which took place at or near the critical speed, was of as large an amplitude as possible without the paddles of the damper coming too close to the walls of the oil channel into which they dipped. This gave the greatest possible magnification of the phenomena to be observed. A similar adjustment was required whenever the mass of the flywheel or the viscosity of the damping fluid was changed.

21 The neon glow lamp was supplied with current from the plate circuit of two 50-watt pliotrons in parallel (see Fig. 9) which

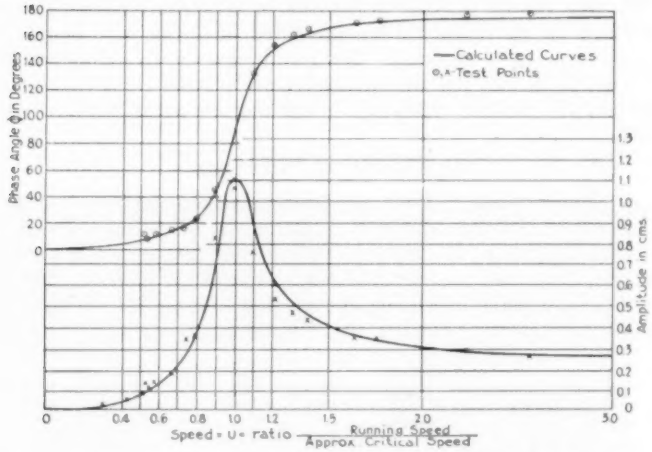
Phase Angle Φ in Degrees
Amplitude in cms
—Calculated Curves
o,x-Test Points
Speed = U= ratio Running Speed / Approx. Critical Speed

FIG. 10 PHASE ANGLE AND AMPLITUDE VARIATION WITH SPEED IN A WHIRLING SHAFT FOR A SMALL AMOUNT OF DAMPING, SHOWING AGREEMENT BETWEEN EXPERIMENTAL AND CALCULATED RESULTS

Equivalent mass, M = 0.683 grams
Elastic constant, c = 341 grams per cm.
Critical speed, ω_c = 3.56 r.p.s.
Eccentricity, e = 0.2268 cm.
Damping factor, b = 3.168 grams per cm. per sec.
Curve factor, a = 0.208

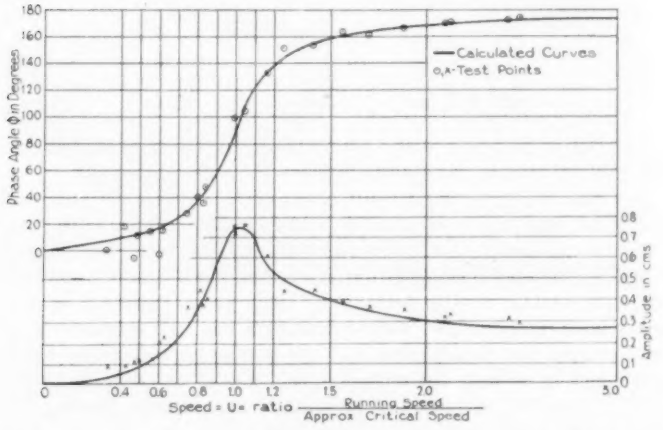
Phase Angle Φ in Degrees
Amplitude in cms
—Calculated Curves
o,x-Test Points
Speed = U= ratio Running Speed / Approx. Critical Speed

FIG. 11 PHASE ANGLE AND AMPLITUDE VARIATION WITH SPEED FOR A SLIGHTLY GREATER AMOUNT OF DAMPING, SHOWING AGREEMENT BETWEEN EXPERIMENTS AND CALCULATED RESULTS

M = 0.685 grams
c = 341 grams per cm.
ω_c = 3.56 r.p.s.
e = 0.2395 cm.
b = 4.99 grams per cm. per sec.
a = 0.3253

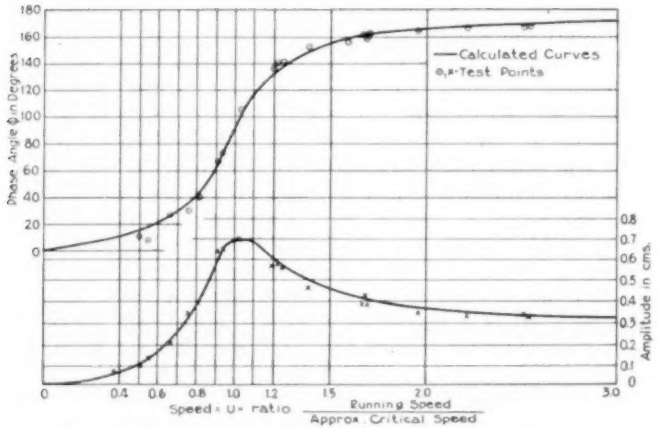

FIG. 12 PHASE ANGLE AND AMPLITUDE VARIATION WITH SPEED FOR HEAVY DAMPING, SHOWING AGREEMENT BETWEEN EXPERIMENTS AND CALCULATED RESULTS

M = 0.685 grams
c = 341 grams per cm.
ω_c = 3.56 r.p.s.
e = 0.287 cm.
b = 6.34 grams per cm. per sec.
a = 0.416

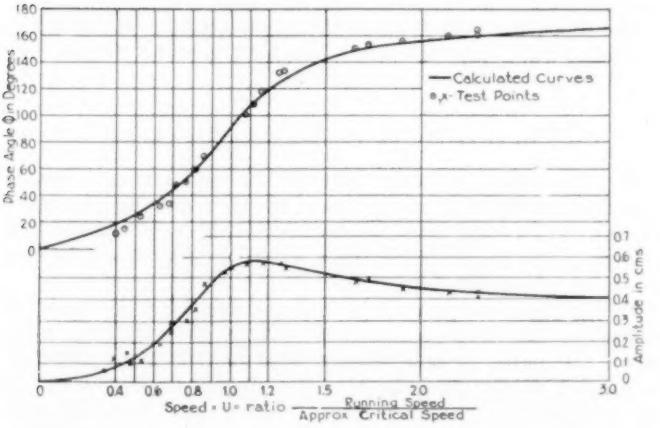

FIG. 13 PHASE ANGLE AND AMPLITUDE VARIATION WITH SPEED FOR VERY HEAVY DAMPING, SHOWING AGREEMENT BETWEEN EXPERIMENTS AND CALCULATED RESULTS

M = 0.685 grams
c = 341 grams per cm.
ω_c = 3.56 r.p.s.
e = 0.364 cm.
b = 10.18 grams per cm. per sec.
a = 0.671

were operated by the contact points R placed at the base of the model shaft and controlling the grid potential. Once each revolution, the grids, normally at a negative potential, were made positive, thus allowing an instantaneous current to flow in the plate circuit. This arrangement was used to avoid sparking at the contact points and the consequent irregular flashing of the lamp.

22 The intermediate jack shaft through which the machine was driven was provided with a flywheel in the final observations. When this was done, besides driving the machine from a storage battery, stable operation was obtained at all speeds, however near the critical. When the speed of the model shaft is raised slowly, the amplitude of whirl of S, the indicator for the shaft, will gradually build up to a maximum at critical speed. On further increasing the speed the amplitude of whirl decreases and approaches at high speeds a value equal to the eccentricity of the model. Below critical speed the eccentric flywheel M appears to wobble around the shaft, whereas above critical speed the flywheel appears to rotate smoothly in space while the shaft whirls around it, as described in Par. 4. This phase change, which takes place at critical speed, is clearly shown in the curves, Figs. 10 to 13.

23 Readings of the phase angle and the amplitude were taken for small increments of speed, from the lowest giving a readable amplitude up to about three times critical, and were then checked by the same number of observations made while reducing the speed, making about twenty readings for each curve obtained.

24 At the end of each set of readings, a record of the amplitude was taken for a very high speed, about six times the critical, from which the eccentricity was calculated.

Computation of Curves and Discussion

25 The critical speed of the shaft for each attached flywheel was obtained both by observation of the maximum amplitude and by calculation following the method of Morley[1] which takes account of the weight of the shaft, as well as that of the flywheel, giving an equivalent value of M.

26 Knowing the critical speed of the model for the conditions of the run and knowing that the phase angle is 90 deg. at this speed, the actual 0- and 180-deg. lines for the phase angle can be located on the phase-angle curve already drawn. These are the experimental curves which are to be compared with the curves calculated from the theory.

27 Using an approximate value for the eccentricity, determined by direct measurement from the apparatus, and using the amplitude r_{sc} at critical speed, an approximate damping constant is obtained from Equation [**9**], M being the equivalent mass.

[1] Arthur Morley, *Engineering*, July 30, 1909, p. 135.

28 Upon substituting in Equation [7] the value of ω and the corresponding value of r_s obtained from the high-speed reading, and the value of b just obtained, and solving for e, a more nearly correct value of the eccentricity can be obtained.

29 An exact value of b can be obtained now by putting this new value of e in Equation [9] and again solving.

30 In several cases the value of b obtained by the above method was checked by use of the ordinary formula for a damped oscillation,

$$y = y_0 e^{-\frac{bt}{2M}} \sin \omega t \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [10]$$

Loading the top of the shaft heavily made it possible to observe the rate of decrease of the amplitude of transverse vibrations and thus determine the damping constant b.

31 The elastic constant of the shaft is obtained from Equation [8], $\omega_c = \sqrt{\frac{c}{M}}$, where ω_c corresponds to the critical speed of rotation of the shaft. Direct measurement verified the value of the elastic constant obtained by this equation.

32 Thus all the constants needed to determine the shape of the curves of phase angle and of amplitude plotted against speed were obtained from observations at easily determined points, namely, at critical speed and at a very high speed.

33 In plotting the curves, by substituting u and a and making use of Equation [6], Equations [4] and [7] were changed to the forms

$$\tan \phi = \frac{ua}{1-\omega^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [11]$$

$$r_s = r_{sc} \sin \phi = \frac{eu}{a} \sin \phi \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [12]$$

where

$$u = \frac{\text{running speed}}{\text{critical speed}} = \frac{\omega}{\omega_c} = \omega \sqrt{\frac{M}{c}} \quad . \quad . \quad . \quad . \quad [13]$$

and

$$a = \frac{\text{eccentricity}}{\text{amplitude of whirl at critical speed}} = \frac{e}{r_{sc}} = \frac{b}{\sqrt{Mc}} \quad . \quad [14]$$

(See Equation [9])

34 From these equations, ϕ and r_s were plotted against u, as shown in Figs. 10 to 13, or what amounts to the same thing, against ω, since u and ω are proportional, as seen from [13]. Thus the curves represent the variations of ϕ and r_s required by theory, using constants determined by careful measurement. The circles and crosses, which indicate the values for ϕ and r_s actually ob-

served as the speed was varied, are seen to fall nearly upon the theoretical curves.

35 The factor a determines the shape of the curves of ϕ and r_s plotted against u. By means of it curves of ϕ and r_s can be directly compared, whether the apparatus be large or small or of high or low critical speed. A small value of a gives curves like those of Fig. 10, where the change in ϕ is rapid and r_s has a decided peak in passing through the critical speed. With a large, curves like those of Fig. 13 are obtained.

36 The damping for the run of Fig. 11 was made relatively greater by increasing the viscosity of the damping fluid, which has the effect of increasing a and flattening the curves.

37 In this curve the displacement past critical speed of the point of greatest amplitude begins to appear, as is required by the theory. This is seen by determining the value of ω for which r_s is a maximum from Equation [7] by the usual method. This is found to be

$$\omega = \frac{2c}{\sqrt{4Mc - 2b^2}} \quad \ldots\ldots\ldots [15]$$

When b is relatively small, this becomes

$$\omega = \sqrt{\frac{c}{M}} = \omega_c$$

The heavy damping in Fig. 13 causes the vibrations to be nearly aperiodic.

38 The agreement between these curves and the test points is very good for the amplitude curve: below critical speed the mean variation is 4 per cent; above critical it is 2 per cent of the critical speed value. The average deviation of less than 2 deg. of the phase-angle points from the theoretical curves is well within the limit of accuracy of the apparatus. In nearly every case the test points are evenly distributed on either side of the calculated curves.

DISCUSSION

PAUL A. HEYMANS.[1] The authors of the paper have presented a simple theory of the action of external frictional forces on the whirling of rotating shafts and have substantiated their theory with very skillful experimenting. The question which immediately has arisen in the writer's mind is whether this theory can be extended to the action of internal frictional forces which are the more fundamental forces to be considered in whirling.

The theory cannot be extended to internal frictional forces in its present form because the equations [1] do not express the

[1] Assistant Professor of Theoretical Physics and Photoelasticity, Massachusetts Institute of Technology, Cambridge, Mass. Assoc-Mem. A.S.M.E.

instantaneous elastic equilibrium at any point of the rotating shaft. Indeed, F_3 should then represent the instantaneous value of the internal frictional force. Writing $F_3 = b\omega r_s$ it would be assumed that the plane of the deflected shaft rotates with the angular velocity ω of the shaft itself around its deflected axis. If this assumption were correct, there would be no forces due to internal friction. Consider the deflected shaft as represented in Fig. 1 of the paper. If the plane of the deflected shaft is rotating with the same angular velocity as the shaft itself around its deflected axis, each line of fibers parallel to the deflected axis of the shaft will remain in the same state of stress. Whereas if the plane of the deflected shaft is not rotating with the same angular velocity as the shaft, only then will each line of fibers alternately be stretched and contracted, and the internal frictional forces due to this alternating strain are those which tend to drag the plane of the deflected shaft with the rotating shaft. It follows that the velocity ω in the relation $F_3 = b\omega r_s$ should be different from the frequency ω of rotation of the shaft around its deflected axis in the relation $F_1 = b\omega^2 r_s$.

The writer would be interested to know if the authors could not, with their present apparatus, measure this velocity of rotation of the plane of the deflected shaft and thereby secure direct information on internal friction itself.

When extending the theory to internal friction the value of the coefficient of friction b could not be assumed to be constant. Let us consider a linear free damped oscillator of mass m. Its dynamic equation of motion is:

$$m\frac{d^2x}{dt^2} + b\frac{dx}{dt} + a^2x = 0 \quad \ldots\ldots [16]$$

where a^2x is the elastic force, $b\dfrac{dx}{dt}$ is the frictional force, and $m\dfrac{d^2x}{dt^2}$ is the inertia force.

The solution of Equation [16] is

$$x = Ax_1 + Bx_2$$

where

$$\left.\begin{aligned} x_1 &= e^{\frac{-b}{2m}t} \times e^{+\sqrt{\frac{b^2}{4m^2} - \frac{a^2}{m}}\,t} \\ x_2 &= e^{\frac{-b}{2m}t} \times e^{-\sqrt{\frac{b^2}{4m^2} - \frac{a^2}{m}}\,t} \end{aligned}\right\} \quad \ldots\ldots [17]$$

and where A and B are two integration constants. The amplitude of the free vibration is an exponentially decreasing value $e^{\frac{-b}{2m}t}$ which will become zero only after an infinite time. In any physical case the oscillation dies out within a finite time, which may be

very short if b is somewhat great. This discrepancy between theory and observation is due to the fact that the frictional constant is not a constant but depends upon the velocity, increasing rapidly once the velocity has fallen below a value v_1 as shown on the diagram Fig. 14. The coefficient of friction b is only constant for values greater than v_1.

As each fiber of the shaft is alternately stretched and contracted with a frequency depending upon the number of revolutions per second of the plane of the deflected shaft and the number of revolutions of the shaft around its deflected axis, its velocity of deformation varies, passing cyclically through zero at the time of maximum tensional and compressional displacements and through a maximum when the deformation goes through its zero value. When writing $F_3 = b\omega r_s$ it is assumed that the frictional

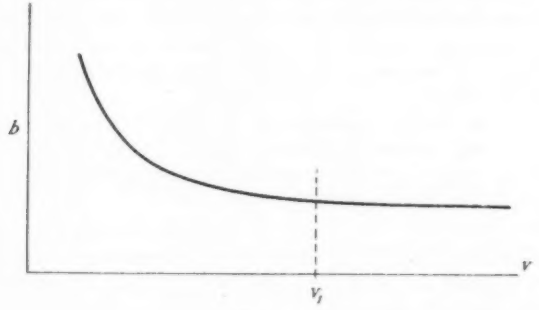

FIG. 14

coefficient b is constant throughout each cycle and for all velocities ω.

In Equations [1], $F_2 = cr_s$, which relates the elastic restoring force to the transverse deflection, could not the factor of proportionality be given an explicit expression? Indeed, the transverse deflection r_s of a shaft which is supported at its end is

$$r_s = \frac{1}{48} \frac{F_2 l^3}{EI}$$

where

l = distance between the supports
E = Young's modulus of elasticity
I = moment of inertia
F_2 = deflecting force which is equal and opposite to the elastic restoring force.

In the case of the apparatus of Fig. 5*a*, a similar relation for the deflection when the shaft extends beyond one of the supports should give the value of the coefficient c. If this is the case, how

does the value calculated that way compare with the values observed as given in Figs. 10 to 13?

It might be beneficial to call attention to the units used by the authors of the paper for the different coefficients given in Figs. 10 to 13. For the equivalent mass, the gram is used as gram-mass in accordance with the units of the c.g.s. system, whereas for the other coefficients the gram is used as gram-force or gram-weight.

The damping coefficient b is being given the dimensions of gram (force) per cm. per sec., and, therefore, should have symbolically the dimensions $\left[\frac{MLT^{-2}}{LT}\right]$ or $[MT^{-1}]$. Indeed, an acceleration is dimensionally $[LT^{-2}]$ and is expressed in cm. per sec. per sec. However, from the relation $F_s = b\omega r_s$ it follows that the dimensions of b must be $\left[\frac{MLT^{-2}}{T^{-1}L}\right]$ or $[MT^{-1}]$. Therefore, the units of b should be either gram-mass per second or gram-force times seconds per centimeter.

The authors state in Par. 37 that the heavy damping in Fig. 13 causes the vibration to be nearly aperiodic. As is well known, a vibration will go over to an aperiodic motion when $\left(\frac{b^2}{4m^2} - \frac{c}{m}\right) = 0$ (using the symbols of the paper), or $b^2 = 4mc$. Using the values for m and c given in Fig. 13 of the paper, we calculate for b the value $b = 31$ against the observed value $b = 10.18$. As the whole range of experimental values of b from a small damping to a very heavy damping only goes from $b = 3.168$ to $b = 10.18$, does not the discrepancy between the calculated value for an aperiodic and the observed value for a very nearly aperiodic motion seem abnormal?

H. D. Taylor.[1] The authors have given credit to the writer for independent use of their analytical method of deriving the equations of shaft behavior which they have verified by experiment. The writer wishes to explain that, before using this method, he was familiar with the results as worked out from differential equations by Jeffcott and Föppl. In short, he knew the answer in advance, and therefore deserves no credit — save perhaps for the short cut.

The subject of shaft behavior at critical speed has been in the past somewhat mysterious, as far as a considerable portion of the literature has been concerned. The usual definition of critical speed has been that speed at which a shaft becomes unstable, its stiffness being completely neutralized by centrifugal forces, so that it becomes incapable of resisting the slightest

[1] General Electric Co., Schenectady, N. Y.

deflecting tendency. While this was a theoretical definition, being based on a simple analysis which neglected friction, it was fairly well borne out by experience — that is, by experience with machines and models which were none too well balanced. The practical result has been that it has become a fixed rule of design to avoid critical speeds, as calculated, by wide margins. That is undoubtedly a wise policy, but not one conducive to familiarity with the actual behavior of shafts at critical speed. Experience with models and with commercial machines has convinced the writer that the vibration encountered at critical speed in general is not evidence of instability so much as of unbalance. For instance, refined balancing of a model rotor, which had been extremely violent even in passing rapidly through critical speed, resulted in such smooth operation that the critical speed could not be detected, however slowly it was passing through. In fact, the speed could be held constant at the exact critical value, as nearly as could be readily measured, without the least tendency for vibration to build up. Furthermore, the addition of measured weights in successive runs, to cause intentional unbalance, produced definite deflections at critical speed, which proved to be approximately proportional to the degree of added unbalance.

Following these experiments, the equations of Jeffcott were called to the attention of the writer by Dr. B. L. Newkirk, with the suggestion that they be compared with the performance of the model. This suggestion was carried out in a special series of tests, using dial indicators to measure the amplitude of shaft vibration, and an improvised stroboscopic device to indicate the "phase angle," or angle of lag, between the heavy side of the rotor and the deflection of the shaft. The model consisted of a uniform horizontal shaft $1\frac{1}{4}$ in. in diameter, carrying a 400-lb. disk wheel midway between two journal bearings which were separated 40 in. center to center. The critical speed was about 1200 r.p.m. No special damping device was arranged for these tests, such damping as there was being incidental to the set-up. The model was quite sensitive to unbalance; an eccentricity of 0.001 in. caused a total amplitude at critical speed of 0.020 to 0.050 in., depending on the lubrication, as will be explained. Some of the results may be outlined as follows:

1 With the same degree of unbalance, the amplitude reached at critical speed decreased as the quantity of oil supplied to the bearings was increased; this indicated the oil film in the bearings as the major source of damping.

2 With constant oil conditions, the "critical amplitude" was definite, repeatable, and proportional to the degree of unbalance. Acceleration in moderate degree had no appreciable effect.

3 The test curves of amplitude and angle of lag vs. speed, when compared with other curves representing the theory, showed good

general agreement as to shape, though some minor differences were apparent.

4 No direct measure of the damping forces considered in the theory being available, a number of different assumptions as to the law governing the variation of damping with speed, amplitude, etc., were introduced into the theory, and the resulting equations were found to check with the tests about as well as those based on the usual assumption (damping proportional to velocity), but no better.

The conclusion drawn was that when all the conditions assumed in the theory are fulfilled, at least approximately, actual shaft behavior agrees quite closely with the theoretical.

E. O. WATERS.[1] The peculiar behavior of an unbalanced rotating shaft has been a matter of common knowledge for many years, and has received increasing attention as reciprocating machinery has been replaced by high-speed rotating machinery throughout the mechanical world. Briefly stated, the most striking phenomena are these:

1 Increasing deflection of the shaft as the speed of rotation increases from zero to the so-called "critical speed."

2 Destructive deflection at the critical speed, unless an external force is introduced, such as the frictional force F_3 of the present paper.

3 Gradual approach to the state of rotation about the center of mass of the shaft, as the speed increases above the critical value.

The authors should be commended for proving by actual observation that these phenomena exist, and that the quantitative relations are identical with what the theory indicates. In reality, their apparatus does the same thing as a balancing machine in which a loaded shaft is rotated first forward, then backward, and the "high spot" marked each time with a scriber. If it were not for the damping force there would be only one high spot, and the heavy side of the shaft would be immediately located.

The authors have assumed that the friction is of a viscous nature, i.e., is directly proportional to the velocity. It is, however, a simple matter to get the same general results with any kind of friction, either constant or a function of the velocity. For example, if we let $F_3 = b\omega^n r_s^n$, then the phase angle ϕ, at critical speed, is 90 deg., and the amplitude,

$$r_s = \frac{1}{\omega_c} \sqrt[n]{\frac{Me\omega_c^2}{b}}$$

This latter expression indicates that an increased intensity of damping, either in the index n or the factor b, will correspondingly

[1] Assistant Professor of Mechanical Engineering, Yale University, New Haven, Conn. Jun. A.S.M.E.

reduce the vibration, and if either one becomes zero, the amplitude increases indefinitely.

Are there other forces besides friction, that possibly can prevent excessive vibration at the critical speed? There seems to be a general impression that a shaft can "safely pass through" this speed if it is not allowed to dally on the way. Jeffcott, in the paper to which the authors referred, shows that a finite time interval is required for building up a large amplitude at the critical speed even when the friction is zero, and suggests that this makes it possible to get through the speed without undue danger to the shaft. More recently, Messrs. Thearle and Dent have endeavored to prove that an angular acceleration impressed on the shaft will have the same general effect as a damping force, i.e., a phase displacement is produced, and the amplitude of vibration is restricted. The results obtained are surprisingly simple, but, unfortunately, certain assumptions and approximations were made which do not seem justified in a rigorous mathematical analysis. It cannot be correct to neglect the radial acceleration of the shaft in comparison with the tangential, when we know for a fact that as a shaft just comes up to a critical speed (damping being zero) the radius of its path is very suddenly increasing, while the velocity of rotation is almost unchanged.

The equations of motion for an unbalanced shaft rotating with *variable* angular velocity are easy enough to set up but not so easy to solve. The writer would like to know whether any rigorous solution has been obtained; also, whether any other action besides that of friction has been found that is capable of restraining the vibration at critical speed. If not, it should be possible to take almost any small shaft, mount it in self-aligning bearings, place it in a vacuum, and speed it up until it breaks.

BURT L. NEWKIRK.[1] The authors have developed the expressions for the radius of whirl and the phase angle of a loaded unbalanced shaft by a simple application of an equilibrium diagram. These expressions are usually derived as part of a solution of the differential equation of motion of a rotating shaft.

The authors have then tested these relations by observation of a rotating mass supported eccentrically at the top of a vertical shaft. The two radial bearings are at the lower end of the shaft, and relatively close together. The proportions of the apparatus are such that, in comparison with rotors of motors, generators, and turbines, the rotor of this model is slender, the unbalance is great, and the damping is great. These features were purposely exaggerated to make the apparatus suitable for convenient comparison with the commonly accepted theory of shaft behavior.

[1] Research Laboratory, General Electric Co., Schenectady, N. Y.

The interesting result of this study is the close agreement between the curves of Figs. 10 to 13, and the results of observation. The experimental work followed a study of the same subject by Mr. H. D. Taylor, as already described.

In interpreting tests of model rotors it is important to remember that rotors of heavy machinery differ in certain fundamentals from models. It is difficult to balance a model rotor relatively as well as a turbine or generator rotor is balanced. Models, especially those intended for quantitative study, are likely to have ball bearings, whereas heavy rotors run in journal bearings have in many cases a length equal to twice the diameter. The bearing supports of model rotors are usually rigid enough to satisfy the assumption of the theory that the supports are entirely rigid. The bearing standards of heavy rotors can hardly be made rigid enough to fulfill this condition. The heavier the rotor, the more likely we are to find that the yield of the support at the bearings when the load is applied is comparable with the deflection of the shaft at its middle. Not only this, but the rigidity of the bearing support is different in different directions.

These are three fundamental differences, and the resulting divergence in shaft behavior is correspondingly great. Because of these circumstances heavy rotors, when well balanced, show only very slight vibration, or none at all, when running at their calculated critical speeds.

With rotors of moderate size, say, of 5000-kw. turbo-generators of flexible-shaft design, we find some vibration near the calculated critical speeds, but resonant vibrations of one sort or another developing at other speeds. These resonant vibrations are usually determined by the elastic characteristics of the bearings or other parts of the structure.

In the case of the largest rotors, bearing elasticity has a predominant effect in determining the critical speed, so that critical-speed phenomena do not occur at all as contemplated in the common critical-speed theory. Here we find vibration in a horizontal plane at one speed and in a vertical plane at a different speed, and possibly other vibrations arising in different parts of the structure at still different speeds. The stimulus to these vibrations is usually the unbalance.

On account of these differences, a point of view somewhat different from that of the common theory of shaft behavior has proved very valuable to the writer. It was suggested by Dr. Moss, of the Thomson Research Laboratory. Chuck a rod in a lathe with no support at the projecting end of the rod, but let it pass through a guard ring so that the rod will not whip out too far when the lathe runs. Now run the lathe, causing the rod to rotate, and set the rod into transverse vibration. It will be found that it will vibrate in a plane, or about as it would if the lathe

were not running. That is, rotation and vibration may be superimposed upon a shaft without mutual interference. This behavior is of course in conformity with the mechanics of elastic materials, but the point of view is usually not emphasized in the theory of shaft behavior. From this point of view the critical-speed phenomena of shafts are simply resonant-vibration phenomena, and by treating them as such we may take into consideration the effect of unbalance, length of bearings, elasticity of bearings, and other circumstances not enumerated above. It would be difficult if not impossible to evaluate the effect of these circumstances on the vibrational behavior of machines, on the basis of the common theory of shaft behavior.

Considered from the point of view outlined above, the calculated critical speed of a machine is the speed at which resonant vibration would develop if the balance were bad, the bearings short and infinitely rigid, and if the machine were run at this speed long enough for resonance to build up. As a matter of fact, however, well-balanced machines do run at their calculated critical speeds, and at all other speeds within limits set by stresses due to centrifugal force, without excessive vibration.

S. TIMOSHENKO.[1] The vibration phenomenon of a shaft while passing through its critical speed is very interesting from a theoretical point of view and has a great practical importance. A comprehensive literature on the subject already exists, and the methods developed for figuring "critical speeds" have a high accuracy. Until now, however, certain points of this phenomenon have not been satisfactorily explained by analysis and recourse to experimental study has become necessary. By using the experimental method, A. Stodola showed the existence of critical speeds of a secondary type and obtained also the effect of damping on the amplitude of vibration of the shaft while running at the critical speed.[2]

In the paper under discussion the conditions of steady motion only are considered. The analysis made coincides with that of Stodola, but the arrangement of experiments is different. In the authors' work a stroboscopic method has been used for obtaining the instantaneous position of the disk and the shaft. This method has already proved useful in the study of many dynamic problems, and in the case under consideration it has yielded results which are in good agreement with the theory.

The writer believes that continuation of this work may give interesting data and may assist in clearing up points which, until

[1] Research Engineer, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa. Mem. A.S.M.E.

[2] See *Schweizerische Bauzeitung*, 1916; also Dampf und Gas Turbinen, 5th Ed., p. 361.

now, have had no satisfactory analytical explanation. It will be interesting, for instance, to obtain by using the stroboscopic method a complete picture of the vibratory phenomena of a horizontal shaft produced by its own weight (critical speed of secondary type). The study of the transient oscillations when the speed of the shaft is going through the critical range may also yield interesting results. It will be important, for instance, to get experimentally the maximum amplitude when different rates of changing the speed of the shaft and different intensities of damping are used while passing through the critical speed.

THE AUTHORS. In his discussion of the paper, Dr. Heymans points out that the damping coefficient, as used, does not take account of the internal friction within the shaft or rotor itself. In answer to this, the authors would say that it would be wrong to include this internal frictional force for the reason that it can arise only when the speed of revolution of the plane of deflection of the shaft and the speed of rotation of the shaft are not the same, that is, in Figs. 1 and 2, where *OS* and *SG* revolve at different angular velocities. In the case investigated, however, for a given speed of rotation, an outside fiber of the shaft, which is in tension, remains in tension, and an inside fiber, which is in compression, remains in compression as the shaft revolves, so that no energy is dissipated due to a working of the material within the shaft. This equality between the whirling or whipping frequency and the rotational frequency is characteristic of all critical-speed vibration phenomena in revolving shafts. There is a type of shaft vibration, however, which is directly excited by such internal friction, but it can arise only when the speed of rotation of the shaft exceeds its whirling frequency. For a discussion of this type of vibration, see the footnote below.[1]

Regarding the assigning of an explicit value to the constant c in Equation [1], this can indeed be done, but the relation is not as simple as that given by Dr. Heymans, because part of the deflection takes place in the portion of the shaft between its bearings, and not all of it in the overhanging portion.

The dimensions of the coefficient b are correct if the expression grams per cm. per sec. is understood as grams $\div$ cm. $\div$ sec., which seems to the authors to be the more natural interpretation, instead of as grams $\div$ cm. $\times$ sec. as interpreted by Dr. Heymans.

The statement of Par. 37 that heavy damping would make the vibration almost aperiodic is, of course, not correct, as Dr. Heymans says. It is only the transient oscillations which would become aperiodic, such as would be observed if the shaft were given an initial deflection and set free to vibrate while not revolving.

[1] A. L. Kimball, *Phil. Mag.*, April, 1925, p. 724. See also B. L. Newkirk, *General Electric Review*, April, 1924, p. 244.

Mr. Taylor describes some of his work on this problem with observations and conclusions, which are of interest because they are made from a comparatively large-sized, high-speed, horizontal model corresponding much nearer to the usual forms of rotating machinery than the rather idealized model used by the authors.

In answer to Mr. Waters' question as to whether a rigorous mathematical solution has been obtained for the case of varying angular velocity while passing through the critical speed, it can be said that no such solution has been obtained so far as the authors are aware. Such a solution is very desirable, and doubtless has been attempted many times.

The experiment mentioned by Dr. Newkirk, which was suggested by Dr. Moss, of observing the vibration of a projecting rod which is revolved in a lathe, has an interesting bearing on the study presented in this paper. The vibration, spoken of as taking place as though the shaft were not running, is the transient oscillation which ultimately disappears, and is omitted in the analysis of this paper just as transient electrical oscillations are omitted in the ordinary treatment of alternating-current circuits. The vibration — due to unbalance — takes place entirely independently of this transient, as can also be seen by an inspection of the equations of motion such as those given by Jeffcott.[1]

[1] H. H. Jeffcott, *Phil. Mag.*, March, 1919, p. 304.

No. 1977

PHOSPHOR-BRONZE HELICAL SPRINGS FROM THE STANDPOINT OF PRECISION INSTRUMENTS[1]

By W. G. Brombacher,[2] Washington, D. C.

Non-Member

This paper gives the results of tests made on phosphor-bronze helical springs investigated at the Bureau of Standards to obtain knowledge useful in the design of springs for precision instruments. The characteristics of the spring material, the method of construction of the springs, the apparatus in which the springs were tested and the procedure followed are set forth by the author. The results relate to stiffness, maximum fiber stress, hysteresis, after-effect, drift, and buckling.

THIS report gives the results of an investigation made at the Bureau of Standards at the request of and financed by the Engineering Division of the Army Air Service in order to study the properties and the design of helical springs. The primary object was to obtain knowledge which would be useful in the design of precision instruments. It was planned to secure a number of sizes of wire of various metals and alloys, to make helical springs of these wires, and to study their performance; however, only phosphor-bronze springs have been constructed and tested. It is hoped that the work can be extended in the near future to include steel springs.

Constants of the Material

2 The phosphor-bronze wire used in this investigation had the composition shown in Table 1, and the constants required in the design of the springs were obtained and are recorded in Table 2.

3 The modulus of torsion was determined by attaching the upper end of a length of wire to a fixed support, fastening symmetrical weights to the lower end, and then obtaining the period

[1] Published by permission of the Director of the National Bureau of Standards of the U. S. Department of Commerce.

[2] Associate Physicist, Bureau of Standards.

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of the system when in rotation about the vertical axis. The relation connecting the constants of the system and the modulus is

$$G = 8\pi Il/a^4P^2$$

in which G is the modulus; I, the moment of inertia of the system; l, the length of the wire in torsion; a, the radius of the wire; and

TABLE 1 ANALYSES OF WIRE

B. & S. gage No.	Copper, per cent	Tin, per cent	Phosphorus, per cent
10	95.8	3.8	0.36
12	95.9	3.6	0.38
14	95.7	3.8	0.35
16	95.6	3.9	0.34
18	95.6	3.9	0.32

Iron: Less than 0.05 per cent in each sample.
Lead, nickel, and zinc: Not detected.

P, the period of the system. The results given above are each the mean of two sets of observations.

4 In addition the modulus was determined for a sample of No. 8 wire by the alternative method of applying a known torque and measuring the corresponding twist. The modulus is computed by means of the familiar relation

$$G = 2Cl/\pi a^4\theta$$

in which G is the modulus in torsion; C, the torque; l, the length of wire under strain; θ, the angle of twist; and a, the radius of the

TABLE 2 MODULUS OF WIRE

B. & S. gage No.	Diameter of wire, in.	Modulus of torsion, lb. per sq. in.
12	0.0812	6.6×10^6
14	0.0647	6.4
16	0.0495	6.5
18	0.0401	6.7
8	0.128	6.5 (estimated)
10	0.103	6.5 (estimated)

wire. A mean of two results gave for G the value 6.34×10^6 lb. per sq. in., which deviates from the assumed value of 6.5×10^6 by 2.7 per cent.

Formulas and Charts for the Design of Helical Springs

5 *Relation Between Stiffness and Number of Turns.* Let

L_D = load in tension or compression on the helical spring
D = deflection of the spring under load L_D
S = stiffness of the spring, defined by the ratio L_D/D
n = number of turns or coils of the spring
a = radius of the spring wire
r = radius of the coils of the spring, taken to the center of the wire
G = modulus of elasticity in torsion
α = pitch angle of the coils of the spring
E = Young's modulus of elasticity.

6 The stiffness of a spring as defined above is restricted in this paper so that it applies only to such deflections as are approximately proportional to the load. Because of this restriction, particular stiffnesses,[1] such as *instantaneous stiffness*, dL_D/dD, or

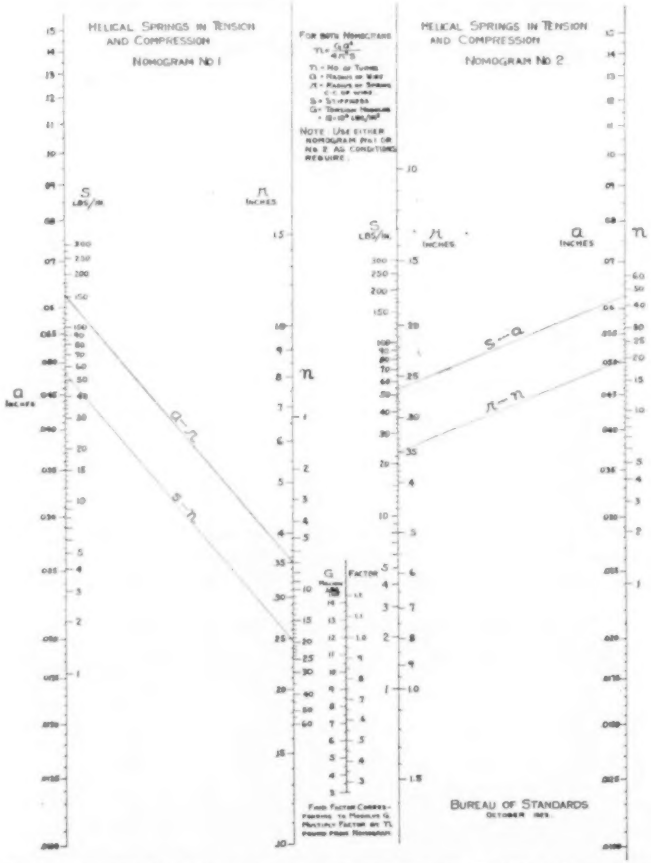
HELICAL SPRINGS IN TENSION AND COMPRESSION
NOMOGRAM NO 1
HELICAL SPRINGS IN TENSION AND COMPRESSION
NOMOGRAM NO 2
BUREAU OF STANDARDS

FIG. 1 NOMOGRAMS NOS. 1 AND 2. DEFLECTION OF HELICAL SPRINGS IN TENSION AND COMPRESSION

initial stiffness, dL_0/dD, have values approximately equal to the stiffness S here defined. The stiffness S, in the sense used here, is also called the effective stiffness.

[1] For a discussion of the various types of stiffnesses, see M. D. Hersey, Diaphragms for Aeronautic Instruments. Nat. Adv. Com. Aero. Tech. Rep. No. 165, 1923.

7 If a force or load is applied so that it only causes a compression or tension of the spring which is symmetrical about the axis of the spring, the relation of the stiffness to the constants of the spring is given by the usual expression $n = Ga^4/4r^3S$, in which it is assumed that α is small and that the changes in the radius of the spring coils for a particular deflection are negligible. This

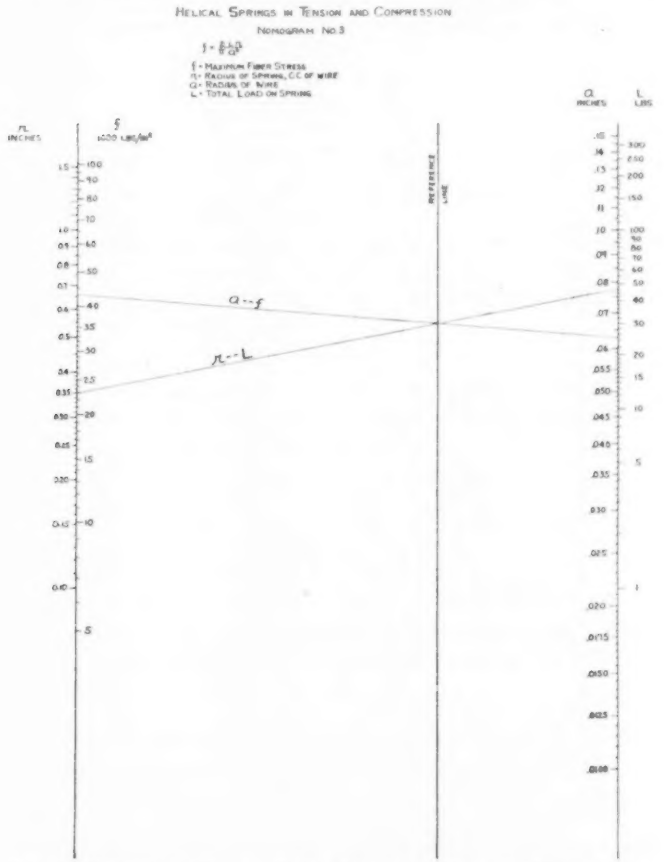

FIG. 2 NOMOGRAM NO. 3. FIBER STRESS OF HELICAL SPRINGS IN TENSION AND COMPRESSION

latter error is commonly thought to be avoided if the ratio of coil to wire diameter is greater than four.

8 *Maximum Fiber Stress.* For most purposes springs should not be loaded beyond the limit at which the deflection is directly proportional to the load, and for all purposes a limit should be

set to the maximum load which can be applied. The maximum load can be determined by a knowledge of the permissible fiber stress in torsion or shear of the spring material used. The maximum fiber stress f of the helical spring under load L is given by

$$f = 2Lr/\pi a^3$$

9 *Nomograms for Use in Design.* Nomograms were drawn up which give solutions for the stiffness and maximum-fiber-stress formula and are shown in Figs. 1 and 2. Nomograms have also been given for this purpose by F. A. Halsey,[1] J. K. Wood,[2] C. Reynal[3] and J. W. Rockefeller, Jr.[4]

Design and Construction of Springs

10 Nine sets of springs were designed with the aid of the nomograms. The details are given in Table 3.

TABLE 3 DESIGN OF SPRINGS

Design No.	Wire No.	Stiffness for 15 turns, lb. per in.	Mean diam. of coils, in.	Load for max. fiber stress of 10,000 lb. per sq. in.
IX	18	3	0.36	0.67
VIII	18	9	0.25	1.0
VII	16	20	0.25	1.9
VI	14	7.6	0.50	2.1
V	12	18.7	0.50	4.4
IV	10	20	0.68	6.3
III	10	6	1.00	4.2
II	8	20	0.92	8.5
I	8	7.6	1.25	6.5

11 *Method of Construction.* An arbor or mandrel was made somewhat smaller than the required inside diameter of the spring. The wire was wound on the arbor which was held in the headstock

TABLE 4 DIMENSIONS OF CONSTRUCTED SPRINGS

Spring No.	Mean diam. of coils, in.	Winding stress, lb. per sq. in.	Spring No.	Mean diam. of coils, in.	Winding stress, lb. per sq. in.
IX-9	0.364	8300	V	0.50	3280
-10	0.363	....	IV	0.68	4150
-11	0.363	....	III	1.00	4250
VIII	0.25	8300	II	0.920	4250
VII	0.25	5400	I-1	1.472	4250
VI	0.50	4600	I-2	1.442	4250

of a lathe. The pitch angle of the coils was determined by feeding the wire to the arbor from the lathe carriage. This was done by hand. The wire was kept under constant tension while being wound by means of weights suspended from the free end which hung over a pulley.

12 The proper arbor diameter and winding loads were found by a cut-and-try method until both straight and accurate springs were obtained. The data on the construction are given in Table 4.

[1] Handbook for Machine Designers and Draftsmen, McGraw-Hill, 1913.

[2] *American Machinist*, vol. 54 (1921), p. 628.

[3] *Arts et Métiers*, Sept., 1922, p. 276.

[4] *Machinery*, 1924, p. 874.

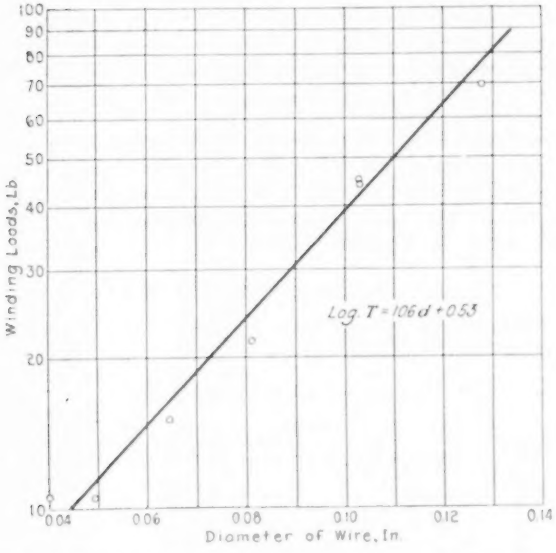

FIG. 3 WINDING LOADS APPLIED IN ORDER TO SECURE STRAIGHT SPRINGS

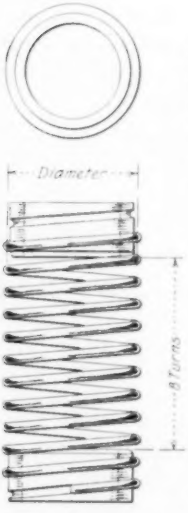

FIG. 4 SPRING WITH END PLUGS

13 *Data of Construction.* Three sets of springs of most of the designs were constructed. The mean diameters of the springs and the stress during winding are given in Table 4. The value of the winding stress is within the proportional limit for phosphor bronze.

14 A relation was found connecting the winding load necessary to obtain a straight spring and the radius of the wire. This is

$$\text{Log } T = 21.2a + 0.53$$

in which T is the winding load in pounds and a the radius of the wire. This relation and the data are plotted in Fig. 3.

15 *Mounting of Springs for Test.* This was done by placing brass plugs at each end of the spring, threaded to the pitch of the springs. See Fig. 4.

Description of the Apparatus

16 The apparatus used to test the springs was developed and built at the Bureau of Standards about three years ago. For the present tests modifications were made, mainly in the method of applying the load. The essential features are shown in Fig. 5. The springs were placed upon a metal plate which was supported by a substantial wooden framework. A flat plate and rod rested on top

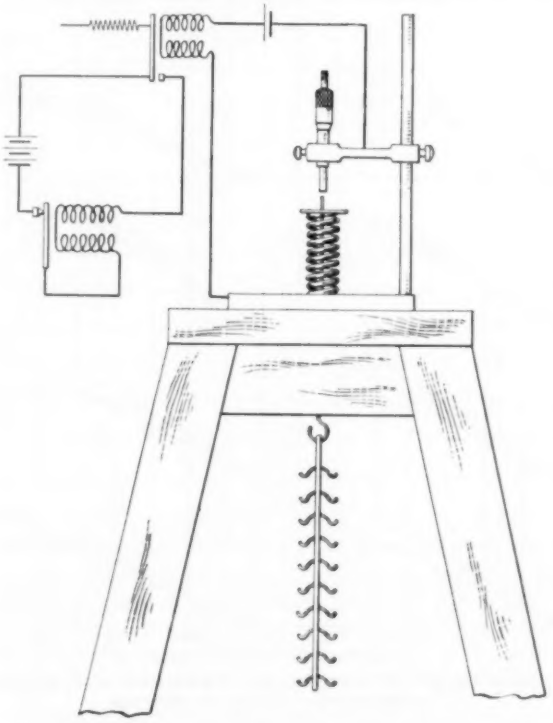

Fig. 5 Spring-Testing Apparatus

of the spring, as clearly shown in Fig. 5. Another rod was suspended from the above-mentioned rod. Weights were hung from the stirrups of the lower rod. These consisted of small paper envelopes filled with lead shot weighed accurately to constitute $\frac{1}{8}$-, $\frac{1}{4}$-, $\frac{1}{2}$-, and 1-lb. loads. A hole punched in the sealed envelope was given a metal rim by pressing into it a patented paper fastener. The weights of the envelopes were adjusted to allow for this procedure. It was found best always to place the loads on the stirrups symmetrically.

17 The deflections were measured by a micrometer head, connected by a steel rod to the base upon which the spring rested. The contact between the top bearing rod and the micrometer was determined by a relay and buzzer system, as shown schematically in Fig. 5. This scheme was adopted in order to reduce the current between the contacts to a minimum.

18 Silver contacts were used, obtained by flowing silver on both the bearing-plate rod and the micrometer tip. The micrometer tip was finished off so as to give a flat surface, while the bearing-plate rod had a hemispherical tip. This obviated the usual difficulties due to oxidation of the contacts since silver oxides are good conductors of electricity. This arrangement was found entirely satisfactory for all tests save those for determining hysteresis.

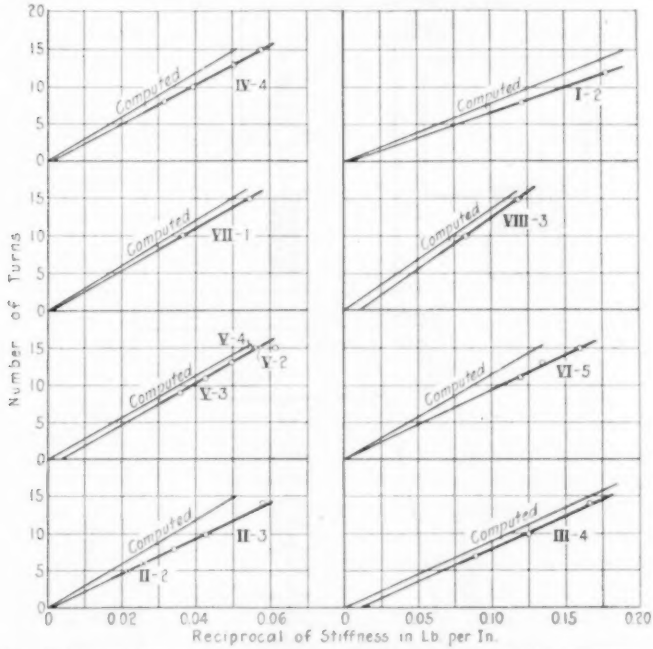

FIG. 6 COMPARISON OF COMPUTED AND EXPERIMENTAL STIFFNESSES OF PHOSPHOR-BRONZE HELICAL SPRINGS

OUTLINE OF TESTS

19 The springs were subjected to loads and the resulting deflections measured. They were tested only in compression. From the load-deflection data information was derived on the stiffness, hysteresis, drift, and after-effect. Information on the load at which the spring buckled was also obtained.

Results of Tests

20 *Stiffness.* The experimental and designed stiffnesses of the springs are fully compared in Fig. 6 and Table 5. The experimental stiffness is from 5 to 20 per cent less than the designed value. This is not due to small values of the ratio of the diameter of spring to diameter of wire, since these values vary from 5 to

TABLE 5 PERFORMANCE OF SIMILAR SPRINGS OF PHOSPHOR BRONZE

Ident. No.	No. of turns	Stiffness, lb. per in.	Computed stiffness, lb. per in.	Average stiffness ratio	r/a	Pitch angle, deg.
IX-9	15	2.54	3.0		9.0	4.5
-10	15	2.70	3.0		9.0	4.5
-11	15	2.66	3.0	0.88	9.0	4.5
VIII-1	15	8.5	9.0		6.2	5.0
-2	15	8.9	9.0		6.2	5.0
-3	15	8.4	9.0	0.95	6.2	5.0
VII-1	15	18.4	20.0		5.0	5.0
-2	15	18.5	20.0		5.0	5.0
-3	15	18.5	20.0	0.92	5.0	5.0
VI-5	15	6.2	7.6		7.7	5.0
-6	15	6.3	7.6		7.7	5.0
-7	15	6.2	7.6	0.82	7.7	5.0
V-2	15	17.7	18.7		6.1	5.0
-3	15	16.4	18.7		6.1	5.0
-4	15	18.2	18.7	0.95	6.1	5.0
IV-2	8	30.7	37.5		6.6	5.5
-4	8	31.6	37.5	0.83	6.6	5.5
III-4	7	11.3	12.9	0.88	9.7	4.5
II-1**	11	16.8	20.4	0.82		
-2	5	47.2	60.0	0.79	7.2	4.0
-3	8	30.0	37.5	0.80	7.2	4.0
-3	5	48.0*	60.0		7.2	4.0
I-1	8.25	7.32	9.00	0.81	11.5	2.4
-2	8	8.25	9.82	0.84	11.2	2.5

* Extrapolated from value for 8 turns. ** Without end plugs.

11.5, 4 being usually given as a safe value. The value of the pitch angle does not seem to be the cause since the discrepancy varies for a constant pitch angle. Part of deviation may be due to an end effect, i.e., due to uncertainty in the number of the coils taking part in the deflection. This is considered in Fig. 6, in which the

TABLE 6 EFFECT OF HEAT TREATMENT ON STIFFNESS

Date of test	Heat treatment, deg. cent.	Increase in stiffness, per cent Spr. No. II-2	Spr. No. IV-4
Feb. 24, 1924	none	0	0
Dec. 27, 1924	none	—3.4	—1.9
Jan. 10, 1925	210*	+2.3	0
Jan. 24, 1925	220**	+2.3	+2.8

* For ½ hour at 210 deg. cent. and 21 hours at 185 deg. cent. ** For 7 hours.

reciprocal of stiffness is plotted against stiffness for the same spring successively shortened after determining the stiffness. Here in three out of eight curves an end effect is indicated, while in five curves no sensible end effect is shown. Two springs were given heat treatment in order to relieve possible stresses set up in construction, the results of which are given in Table 6.

21 *Maximum Fiber Stress.* It was not found possible to put a limiting value to the maximum fiber stress at the proportional limit, because of buckling. Straight-line load-deflection relations were obtained up to 18,000 lb. per sq. in. maximum fiber stress. Handbooks give 20,000 lb. per sq. in. for phosphor-bronze wire in torsion.

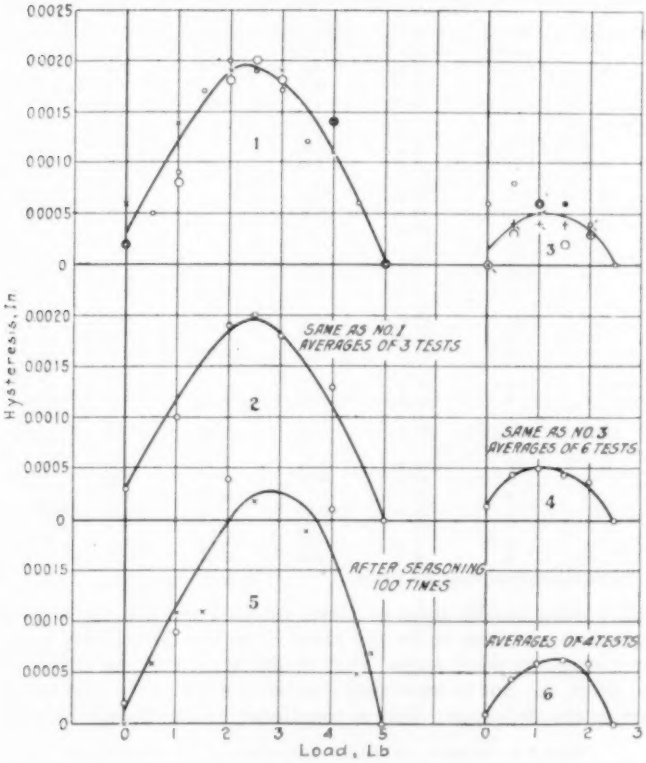

FIG. 7 HYSTERESIS CURVES, PHOSPHOR-BRONZE HELICAL SPRINGS
(No. IV-2; 8 turns.)

22 *Hysteresis.* Since the method of measurement needs some modification before satisfactory results on hysteresis can be obtained, the data given in Figs. 7 and 8 are not considered final. The ordinates of the curves or the hysteresis values are the differences between the deflections at the same load for increasing and decreasing loads. Curves 1 and 2, 3 and 4, and 7 and 8 differ only in that in the first of each pair all the experimental values are

shown, and in the second the experimental values for each load are averaged. The data given are from tests on two similar springs.

23 The data show that the maximum hysteresis depends on the maximum load. Table 7 indicates this more clearly.

TABLE 7

Maximum load, pounds	Maximum hysteresis, inches H_m	Maximum deflection, inches D_m	$\frac{H_m}{D_m}$
2.5	0.0005	0.08	0.006
3.5	0.0010	0.11	0.009
5.0	0.0020	0.16	0.012

24 H_m/D_m is the ratio of the maximum hysteresis to maximum deflection. Here its values plotted against maximum loads give a

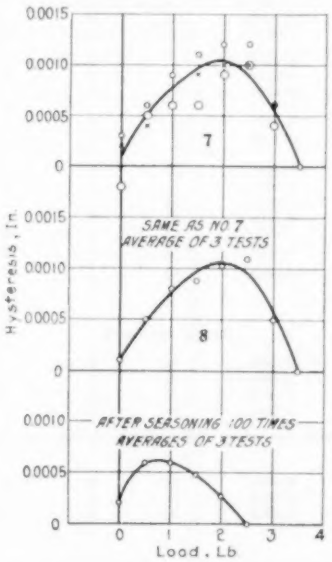

FIG. 8 HYSTERESIS CURVES, PHOSPHOR-BRONZE HELICAL SPRINGS
(No. IV-4; 8 turns.)

straight line which passes through the origin. This is a wholly unexpected result as the ratio is commonly believed to be a constant for loads within the proportional limit. Hence this statement requires further experimental verification. Within the limit of experimental error, seasoning does not seem to affect the amount of the hysteresis.

25 *After-Effect.* The after-effect is defined as the algebraic difference in the readings for zero load at the start and conclusion of a load cycle, the final zero reading being the positive one. The

data are given in Table 8. These indicate conclusively that the after-effect decreases markedly with successive load cycles. The cycles followed one another closely, and were so run that deflection readings were taken at intervals not exceeding two minutes. Thus, for spring I-2 the after-effect decreased from 0.0010 to 0.0007 in. in the second cycle, and to 0.0001 in. for the third cycle.

26 A comparison of the data for spring No. IV-2 before being seasoned and after being seasoned indicates similar reduction in after-effect.

TABLE 8 DATA ON AFTER-EFFECT

(Time of tests as short as possible.)

Spring not seasoned.

Spring No.	Test No.	Maximum deflection, inches	Maximum load, pounds	Number of turns	After-effect, inches	Time of start, p.m.
I-2	1	0.4906	4	8	0.0010	3:20
	2	0.6117	5	8	0.0007	4:12
	3	0.4891	4	8	0.0001	4:33
I-2	1	0.1522	6	6	0.0011	3:11
	2	0.1508	6	6	0.0001	4:02
IV-2	1	0.0813	2½	8	0.0006	1:41
	2	0.0810	2½	8	0	2:02
	3	0.0810	2½	8	0	2:10
	4	0.0809	2½	8	0	2:19
	5	0.0811	2½	8	−0.0002	2:27
	6	0.0807	2½	8	0	3:30
IV-4		0.0792	2½	8	0.0002	

Spring seasoned.

IV-2	1	0.0828	2½	8	0.0007	2:32
Seasoned	2	0.0821	2½	8	0	2:41
200 times	3	0.0822	2½	8	0	2:54
	4	0.0824	2½	8	0	3:02

Note: Compare with preceding data on this spring for unseasoned condition.

						a.m.
IV-4	1	0.0797	2½	8	0.0005	9:34
Seasoned	2	0.0795	2½	8	0	9:43
100 times	3	0.0793	2½	8	0.0002	9:53

27 *Drift.* This quantity is determined with somewhat more certainty than hysteresis or after-effect since the experimental conditions remain constant. The change in the deflection with time under a given load is the drift. The data obtained are given in Table 9 and indicate that the ratio of drift for a given time to the total deflection caused by the load is a constant and approximately amounts to 0.2 per cent for a drift of one hour for the springs of phosphor bronze. The data also indicate that the drift ratio increases with time. If the law that the drift ratio varies as the cube root of the time, as was first suggested by M. D. Hersey, be used to find the drift ratio for one hour from the value for 17 hours which is given for spring No. I-2, the value 0.0015

is obtained. The data are few, but are believed to be trustworthy owing to relatively great freedom from experimental error.

28 *Buckling.* A considerable amount of data on buckling was obtained, as all of the springs buckled even when loaded below the proportional limit. Some of the buckling was caused by the faulty

TABLE 9 DRIFT DATA

Spring No.	Load, pounds	Deflection, inches	Drift, inches	Time, hours	Ratio of drift to deflection	Number of turns
I-2	4	0.489	0.0019	17	0.0039	8
IV-4	8	0.2529	0.00045	1	0.0018	8
	5	0.1604	0.00028	1	0.0017	8

The drifts for spring No. IV-4 were obtained from a curve of the drift-time observations.

centering of the top plate and connecting rod or by the end plugs' failing to be perpendicular to the axis of the spring. These causes were especially evident for long springs of small diameter. A consideration of the problem strictly from the theoretical standpoint

TABLE 10 DATA ON BUCKLING

Spring No.	Number of turns	Criterion B
IX-10	15	3.8
-11	15	4.7
VIII-1	15	6
-2	15	9
-3	15	14
-3	10	10
VII-1	15	10
-2	15	8
-3	15	6.5
VI-5	15	9
-5	13	8
-6	15	9
-7	15	5.5
V-3	13	9
-3	11	9
-3	9	8
-4	15	7
IV-2	8	7
-2	8	8
-4	15	15
-4	13	19
-4	10	19
-4	8	9
-4	8	8
III-4	14	15
-4	10	26
-4	7	13
II-3	14	28
-3	10	32
I-2	8	16

would necessitate the ruling out of data in which the above causes seemed to operate, but practically these causes cannot be eliminated. Table 10 gives the data on buckling. Most of the buckling loads are for the case where the spring buckled decisively, but in some cases it is believed that a shift of the position of the top plate

would have eliminated the buckling. This would not be true buckling. The effect of this accidental cause can be estimated by noting the variation in the buckling loads for groups of similar springs of the same number of turns.

29 Hurlbrink[1] has developed a formula for determining the buckling loads of helical springs in compression. This is, for wire of circular cross-section,

$$B = 45 \frac{r^2}{hD}$$

in which $B \geq 6$ and h = the height of the spring when not under load, D = the deflection under a given load, r = the radius of the coils, and B = the buckling criterion. The idea is that buckling will not occur if B is numerically equal to or is greater than 6. Values of B have been computed for the data available. These are given in Table 10. It is peculiar that the value for B is somewhere near 6 for the springs of small radius, groups V to IX, for which the data were thought to be somewhat uncertain, while for the remaining groups it deviates widely from the value 6. The values for springs IV-2 and IV-4 with 8 turns are the most reliable, since these springs were selected for repeated tests, and here the values of B are both consistent and average well with those found for spring groups IX to V.

30 *Miscellaneous Performance Factors.* It is well known that the stiffness of springs is affected by temperature changes. In these tests the temperature was considered constant. The range of the room temperature from test to test was from 21 to 27 deg. cent., and usually varied less than 2 deg. during any one test.

31 The seasoning of some of the springs has been mentioned. This consisted of deflecting the springs by hand for the stated number of cycles. The deflections corresponded to a maximum fiber stress of approximately 15,000 lb. per sq. in. in two cases and 20,000 lb. per sq. in. in the other case. Some of the effects of seasoning have already been considered. One further effect may be mentioned, which is the effect on the stiffness, data for which are given in Table 11. A slight decrease in stiffness with seasoning is indicated.

TABLE 11 EFFECT OF SEASONING ON STIFFNESS

Spring No.	Stiffness unseasoned, lb. per in.	Stiffness seasoned, lb. per in.	Number of cycles
I-2	8.25	8.15	210
IV-2	30.7	30.0	200
IV-4	31.6	31.4	100
IV-4	31.6	31.4	800

[1] E. Hurlbrink. Berechnung zylindrischer Druckfedern auf Sicherheit seitliches Ausknicken (Theory of Buckling of Helical Springs). *Zeit. des Vereines deutscher Ingenieure,* vol. 54 (1910), p. 133.

DISCUSSION

J. M. LESSELLS.[1] This paper gives some new data on the testing of delicate springs which are interesting. It is of special interest to the writer because an investigation is at present being conducted by Mr. P. L. Irwin of the Mechanics Section of the Westinghouse Research Laboratory on hairsprings used in electrical instruments. As regards the apparatus used, it is our opinion that the relay and buzzer system would not be sufficiently sensitive, and this fact may, in large measure, account for the peculiar hysteresis values obtained. In our original work a "B" battery ($22\frac{1}{2}$ volts) was connected in series with a 60,000-ohm resistance and a very sensitive galvanometer pointer. This arrangement eliminates the contact troubles discussed by the author and was originally used by us not only to measure the spring deflections, but even the extensions of the wire composing the spring. This latter, it must be conceded, is a much more difficult problem. It has since been replaced by another method which for the purposes of measuring wire extensions is more quickly performed. For spring deflections, however, the electrical method is to be preferred. Possibly, the author may be able to obtain further hysteresis data with the modification as suggested.

A. H. MEARS.[2] This paper places at the disposal of the designer of precision instruments nearly all the necessary data on helical springs. An illustrative example or two of the use of the nomograms would be a desirable addition to the paper.

The subject of method of construction is important. That the winding stress is greater for small-diameter wires than for large-diameter wires is checked to a large extent by shop experience. This may be partly accounted for by the fact that in most cases small wires are worked more in drawing than are the large ones.

It seems unfortunate that the project called only for the investigation of phosphor-bronze helical springs for the following reasons:

a The elastic limit of phosphor bronze being much lower than that of spring steel, the problem of designing a spring with a sufficiently small fiber stress to insure small elastic errors is rendered unnecessarily difficult. This difficulty is increased by the limitation of space in which the spring is to be placed.

b The metallurgy of spring steel is far better known than that of phosphor bronze.

c It is not necessary to use a non-magnetic material except in very special cases, and then a spiral type is usually preferable.

[1] Engineer in Charge Mechanics, Research Laboratory, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa. Mem. A.S.M.E.

[2] Research Engineer, Leeds & Northrup, Philadelphia, Pa.

d The temperature coefficients for spring steel are generally smaller than for phosphor bronze. More experimental work has been done upon the temperature coefficients of spring steel.

For these reasons the work should be extended in the near future to include steel springs.

It is the writer's experience that the vibration method for determining experimentally the stiffness of springs gives more accurate results with far less effort than the loading method. It requires only the simplest of apparatus, such as one or two weights and a watch. A comparison of these two methods checked against the theoretical stiffness would be very interesting.

E. J. Loring.[1] Referring to Par. 31, relating to the seasoning of springs, the writer has had an experience indicating an effect contrary to that noted. Some twenty-five years ago, he was engaged in the manufacture of an automobile speedometer of the centrifugal type, in which the calibrating spring was a fairly heavy bronze hairspring. Instruments reaching the shop after two or three years of service showed an increase of about 2 per cent in the strength of the spring. The explanation evolved at that time was that the additional cold-working due to the vibration of the car had stiffened the spring.

In reference to the remarks on drift, is there anything in the tests that would indicate a safe limit of stress for bronze springs that are to stand long storage in a stressed condition? The springs in mind are those used in artillery fuses, some of which are expected to operate within fairly close limits.

The Author. Mr. Lessells' objection to the use of a relay and buzzer system in making the hysteresis measurements is shared by the author mainly on the basis of the scattering of the observations in curves 1 and 3 of Fig. 7. However, it should be stated that the apparatus functioned satisfactorily in all other respects. It is planned to use an optical method for indicating the deflections in further work, especially with steel springs, since their hysteresis is no doubt smaller than that found for the phosphor-bronze springs in this investigation.

The views of Mr. Mears are shared by the author, especially as to the desirability of securing similar data on steel springs. It should be here acknowledged that this work on springs was originally planned in association with Mr. Mears and that the essential parts of the spring-testing apparatus were designed and constructed under his direction.

The question relating to the safe limit of stress for bronze springs when under continual stress which was brought up by

[1] Ordnance Engineer, Washington, D. C. Mem. A.S.M.E. Deceased, Jan. 19, 1926.

Mr. Loring cannot be answered by the author. The best results would undoubtedly be obtained if a bronze of the proper composition were used and properly treated in the manner suggested by B. W. St. Clair.[1]

In closing, it is desired to call attention to the general subject of the proper nomenclature for the elastic properties of springs. In this paper the elastic properties have been defined as hysteresis, after-effect, drift, and stiffness. These properties have received various names in current literature and frequently different properties have received the same name; for example, the first three of the above-named properties have been called hysteresis. This confusion in definition is due largely to the fact that there is no standard method of making the tests which evaluate these quantities. The rate of loading the springs must form an essential part of the definition. The nomenclature followed here, while open to modification, is offered as a basis for consideration of the entire subject.

[1] Springs for Electrical Measuring Instruments, *Mechanical Engineering*, vol. 47 (1925), p. 1057.

No. 1978

FORMULAS FOR THE DESIGN OF HELICAL SPRINGS OF SQUARE OR RECTANGULAR STEEL

By C. T. Edgerton,[1] New York, N. Y.

Non-Member

Although helical springs of square- or rectangular-bar steel are not common, problems involving certain extreme requirements are occasionally encountered which can be met only by such types of springs. The author points out the lack of formulas for calculating any except springs of square-bar steel and then develops formulas for rectangular-bar steel based on the work of St. Venant. For the solution of these formulas he gives tabulated values for two variables which depend on the ratio of the bar's cross-sectional dimensions. An appendix contains four examples in which the application of the formulas is illustrated.

HELICAL springs of rectangular steel are not in common use in this country. They are more expensive to manufacture than round-bar springs, particularly if the section is wide and thin, in which case special fabricating appliances are often required. Spring manufacturers, as a rule, do not encourage their use, and do not carry a variety of sizes of rectangular steel in stock.

2 The square-bar spring, which can be considered as a special case of the rectangular, is popular for certain applications such as safety-valve work, and is occasionally used in general machinery construction, but seldom for locomotive and car suspension and draft rigging. In the latter field the round-bar spring reigns supreme.

3 However, problems involving certain extreme requirements are occasionally encountered which can be met only by a spring of rectangular steel; and every competent spring designer should be armed with correct and convenient formulas for calculating springs of this type. So far as the author knows, no such formulas are available in the standard books of reference.

[1] Manager, Bureau of Statistics, Crucible Steel Company of America.

Contributed by the Special Research Committee on Metal Springs and presented at the Annual Meeting, New York, November 30 to December 4, 1925, of The American Society of Mechanical Engineers.

4 Kent's Handbook gives the following for square-bar springs:

$$P = \frac{0.471Sb^3}{M} \qquad F = \frac{4.712PM^3}{Gb^4}$$

where F is compression (or extension) of one complete turn of the helix under load P, S is the maximum torsional stress in the section, b is the size of section, M is the pitch diameter of the helix, and G is the torsional or shearing modulus.

5 These formulas are developed from the fundamental equations for torsional stress and strain in a bar of square section as found in most textbooks on mechanics. It is generally known by engineers that they are inaccurate. They are based on the assumption that the section retains its original shape when subjected to torsion. The assumption is true for a circular section, the stresses being in complete symmetry around the neutral axis, but in a square or rectangular bar the stresses are not in complete symmetry, and the assumption is false. The reader can observe the distortion for himself by twisting a rubber eraser of the usual rectangular section.

6 The inaccuracies in these fundamental equations for torsion are perhaps not very serious as applied to most machine members, where the torsional movements are small, and the factors of safety ample, but in spring design these conditions are exactly reversed. The torsional movements are relatively large, the springs are often so designed as to be repeatedly stressed nearly to the elastic limit of the material, and the errors in the equations are quite serious.

7 For the more general case of rectangular sections, Kent refers to the well-known formulas given in Reuleaux's Constructor, as follows:

$$P = \frac{2}{3} \cdot \frac{S}{M} \cdot \frac{b^2h^2}{\sqrt{b^2+h^2}}$$

$$F = \frac{3\pi PM^3}{G} \cdot \frac{b^2+h^2}{b^3h^3}$$

the dimensions of the section being $b \times h$. The notation in these equations has been changed slightly to conform to the Kent equations.

8 Merriman, in his work on the mechanics of materials, offers a load equation for the rectangular case, as follows:

$$PR = \frac{2}{9} Sbh^2$$

This is for torsion in general; the equivalent spring formula is

$$P = \frac{4}{9} \frac{Sbh^2}{M}$$

Merriman does not give any formula for the deflection.

9 Both the Reuleaux and the Merriman formulas appear to be attempts partially to compensate, in an empirical way, for the errors in the usual theory.

10 The true formulas for torsional stress and strain in rectangular and other prisms were developed by the French mathematician, St. Venant, by an application of the general theory of elasticity to the case; and were first published in his Memoires des Savants Étrangers about the year 1850. The differential equations, with their transformations, are given in full in Thomson's and Tait's Natural Philosophy. An abridged statement of the solution for rectangulars will be found in Professor Burr's textbook on mechanics, and the Encyclopedia Brittanica article on elasticity contains an extensive reference to the problem. The reader is referred to these sources for an account of the method. The final equations, in the more convenient form given by Professor Burr, are—

For the stress:

$$S = G\alpha h\left[1 - \frac{8}{\pi^2}\left(\frac{1}{\cosh\frac{\pi b}{2h}} + \frac{1}{3^2\cosh\frac{3\pi b}{2h}} + \frac{1}{5^2\cosh\frac{5\pi b}{2h}} + \cdots\right)\right]$$

For the torsional moment:

$$\text{Mom.} = G\alpha bh^3\left[\frac{1}{3} - 0.210083\,\frac{h}{b} + 0.209137\,\frac{h}{b}\left(\frac{1-\tanh\frac{\pi b}{2h}}{1} + \frac{1-\tanh\frac{3\pi b}{2h}}{3^5} + \frac{1-\tanh\frac{5\pi b}{2h}}{5^5} + \cdots\right)\right]$$

where S = maximum stress
b = width of section
h = thickness of section (smaller than b)
G = torsional modulus
α = angle of torsion

11 A brief account of the method by which these rather formidable equations can be reduced to forms convenient for spring design may be of interest.

12 Examination of the formulas will disclose (1) that the dimensions of the section, b and h, appear in the two infinite series only as a ratio, not independently; and (2) that the infinite series in the moment formula is highly convergent, so that all terms

beyond the first can be neglected. This formula can then be simplified as follows:

$$\text{Moment} = G\alpha bh^3\left[\frac{1}{3} - 0.210083\,\frac{h}{b} + 0.209137\,\frac{h}{b}\left(1 - \tanh\frac{\pi b}{2h}\right)\right]$$

$$= G\alpha bh^3\left[\frac{1}{3} - 0.000946\,\frac{h}{b} - 0.209137\,\frac{h}{b}\tanh\frac{\pi b}{2h}\right]$$

Putting $k = h/b$

$$\frac{\omega}{2} = k\left[1 - \frac{8}{\pi^2}\left(\frac{1}{\cosh\dfrac{\pi}{2k}} + \frac{1}{3^2\cosh\dfrac{3\pi}{2k}} + \frac{1}{5^2\cosh\dfrac{5\pi}{2k}} + \cdots\right)\right]$$

$$\frac{\theta}{4} = \frac{k^3}{3} - 0.000946k^4 - 0.209137k^4\tanh\frac{\pi}{2k}$$

we have

$$S = G\alpha bk\left[1 - \frac{8}{\pi^2}\left(\frac{1}{\cosh\dfrac{\pi}{2k}} + \frac{1}{3^2\cosh\dfrac{3\pi}{2k}} + \frac{1}{5^2\cosh\dfrac{5\pi}{2k}} + \cdots\right)\right]$$

$$= \frac{G\alpha b\omega}{2}$$

$$\text{and Moment} = G\alpha b^4\left[\frac{k^3}{3} - 0.000946k^4 - 0.209137k^4\tanh\frac{\pi b}{2k}\right]$$

$$= \frac{G\alpha b^4\theta}{4}$$

13 In the development of the usual helical-spring formulas as given in works on mechanics it is shown that

$$\text{Moment} = \frac{PM}{2} \quad \text{and} \quad \alpha = \frac{2\Delta}{\pi M^2}$$

where P = compressive (or extensive) load on the spring
M = mean or pitch diameter of helix
Δ = deflection of one complete turn of the helix, under load P.

14 Making the substitutions, we get

$$S = \frac{G\Delta b\omega}{\pi M^2} \quad \text{or} \quad \Delta = \frac{S\pi}{G}\cdot\frac{M^2}{b\omega}$$

$$\frac{PM}{2} = \frac{G\Delta b^4\theta}{2\pi M^2} \quad \text{or} \quad P = \frac{G\Delta b^4\theta}{\pi M^3} = \frac{Sb^3}{M}\,\frac{\theta}{\omega} = \frac{Sb^3\rho}{M}$$

where $\rho = \theta/\omega$

15 If S be taken as representing the maximum allowable stress, Δ must then be the maximum safe deflection per turn, or if the

spring is to stand a "solid" test (be compressed tight), Δ will be the maximum allowable space between turns.

16 The two equations

$$P = \frac{Sb^3\rho}{M} \quad \text{and} \quad \Delta = \frac{S\pi}{G}\,\frac{M^2}{b\omega}$$

have been given these particular forms because the latter seem to be generally applicable to helical springs made of any regular section of bar. For example, they are correct for round-bar springs, and in that case the constants have the values

$$\rho = \pi/8 \qquad \text{and} \qquad \omega = 1$$

17 For rectangular sections, the next step is to evaluate ρ and ω for a graded series of values of $k = h/b$. This can be done by the use of the well-known identities

$$\cosh x = \frac{e^x + e^{-x}}{2} \quad \text{and} \quad \tanh x = \frac{e^x - e^{-x}}{e^x + e^{-x}}$$

whence we have

$$\frac{\theta}{4} = \frac{k^3}{3} - 0.000946k^4 - 0.209137k^4\,\frac{e^{\frac{\pi}{2k}} - e^{-\frac{\pi}{2k}}}{e^{\frac{\pi}{2k}} + e^{-\frac{\pi}{2k}}}$$

$$\frac{\omega}{2} = k\left[1 - \frac{8}{\pi^2}\left(\frac{2}{e^{\frac{\pi}{2k}} - e^{-\frac{\pi}{2k}}} + \frac{2}{3^2\left\{e^{\frac{3\pi}{2k}} + e^{-\frac{3\pi}{2k}}\right\}} + \frac{2}{5^2\left\{e^{\frac{5\pi}{2k}} - e^{-\frac{5\pi}{2k}}\right\}} + \cdots\right)\right]$$

18 Following is a specimen calculation for $k = 0.75$.

$$\frac{\pi}{2k} = 2.0944 \qquad \frac{k^3}{3} = 0.14063$$

$$e^{\frac{\pi}{2k}} = 8.1204 \qquad 0.000946k^4 = 0.00030$$

$$e^{-\frac{\pi}{2k}} = 0.1231 \qquad \tanh\frac{\pi}{2k} = 0.97013$$

$$\frac{1}{\cosh\frac{\pi}{2k}} = 0.24262 \qquad 0.209137k^4 \tanh\frac{\pi}{2k} = 0.06420$$

$$e^{\frac{3\pi}{2k}} = 535.48 \qquad \theta = 0.30452$$

$$e^{-\frac{3\pi}{2k}} = \text{insignificant} \qquad \rho = \frac{\theta}{\omega} = 0.25282$$

$$\frac{1}{3^2\cosh\frac{3\pi}{2k}} = 0.00041$$

The third term of the series is insignificant, hence

$$1 - \frac{8}{\pi^2}\left(\frac{1}{\cosh\frac{\pi}{2k}} + \frac{1}{3^2\cosh\frac{3\pi}{2k}}\right) = 0.80301$$

and

$$\omega = 1.2045$$

19 In Table 1 the values of ρ and ω were worked out for values of k from 1.00 to 0.10 by steps of 0.05, and intermediate values by steps of 0.01 were then obtained by the use of the Besselian formulas for interpolation to fifths.

20 Before proceeding to illustrate the use of these formulas, several important conditions must be given consideration, first of which is the allowable fiber stress. The time-honored design figure for round-bar springs is 90,000 lb. per sq. in. But for square bars this can be increased, and for thin rectangular sections it can be very greatly increased. A proof of these statements involves a discussion of the theory of torsional overstrains, which would take too much space at present and must be reserved for a later paper. It can only be pointed out here that the practice, long established among spring makers, of subjecting helical compression springs to an excessive amount of strain (deflection) in their proof tests, results in the introduction of intrasectional stresses — "entrapped stresses" as they have been aptly termed — into the bar. One outward and visible manifestation of these intrasectional stresses is an elevation of the *apparent* elastic limit; another is a degradation of the *apparent* torsional modulus. Finally, owing to the curious distribution of stress in a rectangular bar under torsion, which is developed by St. Venant as a most interesting corollary of his torsion theory (the points of maximum stress are the middle points of the long sides), it is found that the possible elevation of elastic limit by overstrain is greater in a square bar than in a round one, and again greater in a thin rectangular than in a square one.

21 Here a word of caution to the optimistic designer: A high degree of initial overstrain has a rather disastrous effect on the anti-fatigue qualities of a spring, especially if its service involves repeated loads running up close to the capacity. For such purposes the design stress should be conservative. Maximum obtainable stresses should be considered only when the average load on the spring is much below its full capacity, and varies (normally) within narrow limits.

22 Another important fact to be considered is that a square or rectangular bar, when wound hot into a helical spring, suffers a slight deformation of section, its thickness increasing on the side in contact with the mandrel, and decreasing on the outer face, so that the resulting section after coiling is approximately a "keystone shape." This deformation does not seriously affect the

elastic characteristics of the section when under stress, but it does seriously affect the solid or closed height of the spring, and therefore must be considered in designing.

23 Let us assume that the face of the bar in contact with the mandrel retains its original area after coiling; that is,

$$Mh = N(h+u) = (M-b)(h+u)$$

where M is the pitch diameter of the helix, N is the diameter of

TABLE 1 VALUES OF ρ AND ω

k	ρ	ω	k	ρ	ω
1.00	0.4163	1.351	0.55	0.1458	0.998
0.99	0.4092	1.347	0.54	0.1411	0.985
0.98	0.4022	1.342	0.53	0.1365	0.972
0.97	0.3953	1.338	0.52	0.1319	0.958
0.96	0.3883	1.334	0.51	0.1274	0.944
0.95	0.3814	1.329	0.50	0.1230	0.930
0.94	0.3746	1.324	0.49	0.1186	0.916
0.93	0.3678	1.319	0.48	0.1143	0.901
0.92	0.3610	1.314	0.47	0.1101	0.886
0.91	0.3543	1.309	0.46	0.1059	0.871
0.90	0.3476	1.304	0.45	0.1018	0.856
0.89	0.3409	1.298	0.44	0.0978	0.840
0.88	0.3343	1.293	0.43	0.0939	0.824
0.87	0.3277	1.287	0.42	0.0900	0.808
0.86	0.3212	1.281	0.41	0.0862	0.791
0.85	0.3148	1.275	0.40	0.0825	0.774
0.84	0.3084	1.269	0.39	0.0788	0.757
0.83	0.3020	1.263	0.38	0.0752	0.740
0.82	0.2956	1.256	0.37	0.0717	0.723
0.81	0.2893	1.249	0.36	0.0682	0.705
0.80	0.2831	1.242	0.35	0.0649	0.687
0.79	0.2770	1.235	0.34	0.0616	0.669
0.78	0.2709	1.228	0.33	0.0583	0.651
0.77	0.2648	1.220	0.32	0.0552	0.632
0.76	0.2588	1.212	0.31	0.0521	0.614
0.75	0.2528	1.204	0.30	0.0491	0.595
0.74	0.2469	1.196	0.29	0.0462	0.576
0.73	0.2410	1.188	0.28	0.0433	0.557
0.72	0.2352	1.180	0.27	0.0406	0.537
0.71	0.2295	1.171	0.26	0.0378	0.518
0.70	0.2238	1.162	0.25	0.0352	0.499
0.69	0.2182	1.153	0.24	0.0327	0.479
0.68	0.2126	1.143	0.23	0.0302	0.459
0.67	0.2071	1.133	0.22	0.0278	0.439
0.66	0.2017	1.123	0.21	0.0255	0.420
0.65	0.1963	1.113	0.20	0.0233	0.400
0.64	0.1910	1.103	0.19	0.0212	0.380
0.63	0.1857	1.092	0.18	0.0192	0.360
0.62	0.1805	1.081	0.17	0.0172	0.340
0.61	0.1753	1.070	0.16	0.0154	0.320
0.60	0.1702	1.059	0.15	0.0136	0.300
0.59	0.1652	1.047	0.14	0.0119	0.280
0.58	0.1603	1.035	0.13	0.0104	0.260
0.57	0.1554	1.023	0.12	0.0089	0.240
0.56	0.1505	1.011	0.11	0.0075	0.220
			0.10	0.0062	0.200

mandrel, and u is the increase in thickness, or " upset." Then

$$h+u = \frac{Mh}{M-b}$$

and $$u = \frac{Mh}{M-b} - h = \frac{bh}{M-b}$$

or for square bars $$u = \frac{b^2}{M-b}$$

24 Actual measurements of square-bar springs show that the upset u is never as great as this theoretical value. The actual upset varies somewhat with the coiling temperature and other conditions, but in average practice is about three-tenths of the theoretical value, or

$$u = \frac{3}{10}\,\frac{b^2}{M-b}$$

In general the same ratio can be safely assumed for rectangulars, that is,

$$u = \frac{3}{10}\,\frac{bh}{M-b}$$

although if a maximum solid height must be rigidly met, in a rectangular bar design, it would be well to check the estimated upset by an actual trial.

25 Finally, it must be remembered that the end coils of a compression spring, which are squared to form a flat bearing, represent dead or inactive height. If the ends are ground after coiling, as is the usual practice with squares and rectangulars, the inactive height will be equivalent to about one and one-half times the (upset) thickness of the bar, or

$$i = 1\tfrac{1}{2}(h+u)$$

which must be deducted from the total height of spring to get the actual working length.

26 There are certain minor refinements of design which sometimes have to be considered in very exact work, such as the scale loss in heating the bar to coil it, the taper of the mandrel, the shrinkage in pitch diameter on cooling, etc. These can usually be neglected.

27 Let us now assemble the various formulas and methods brought out in the above discussion.

Notation

b = width of section
h = thickness of section } b must be greater than h; see below
M = pitch diameter of spring
H = free length of spring
L = length of spring under load P
E = solid length of spring
i = dead or inactive length
p = pitch of helix, unloaded

n = number of active turns in helix
B = bar length to make the spring, before tapering
C = capacity, or load required to close spring solid
P = working load.

Design Formulas

28 Such of the following formulas as have not already been discussed are self-evident. They are given in the sequence in which they are usually employed, in the checking of an actual design.

$$k = h/b$$

$$u = \frac{3}{10}\,\frac{bh}{M-b}$$

$$i = 1\tfrac{1}{2}(h+u)$$

$$C = \frac{Sb^3\rho}{M}$$

$$\Delta = \frac{S\pi}{G}\cdot\frac{M^2}{b\omega}$$

$$p = \Delta + h + u$$

$$n = \frac{H-i}{p}$$

$$E = n(h+u)+i$$
$$= (n+1\tfrac{1}{2})(h+u)$$

$$\frac{H-L}{H-E} = \frac{P}{C} \text{ or}$$

$$L = H - \frac{P}{C}(H-E) \text{ or}$$

$$P = C\,\frac{H-L}{H-E}$$

$$B = \pi M(n+1\tfrac{1}{2})$$

(Note: The two infinite series in the St. Venant formulas are convergent, in the notation used, only where b is greater than h; therefore the solution is correct only for this condition. If h is greater than b, the two symbols must be interchanged in the formulas for k, C, and Δ, thus:

$$k = b/h \qquad C = \frac{Sh^3\rho}{M} \qquad \Delta = \frac{S\pi}{G}\,\frac{M^2}{h\omega}$$

the other formulas remaining as already written. But such a spring, in which the thickness of section is greater than the width, is very rare indeed.)

APPENDIX

APPLICATIONS OF THE FORMULAS

29 A few examples of the use of these formulas will probably be more illuminating than an abstract discussion.

Example 1

30 A spring of rectangular steel, 1 in. by ½ in., coiled on edge, pitch diameter 4 in., free length 10 in. Determine the solid height, bar length required to manufacture, and the height under a load of 2500 lb. The service is not very severe, and the apparent maximum fiber stress can

run up to 125,000 lb. per sq. in. The apparent torsional modulus will then be about 10,000,000. From the formulas and table:

$$k = h/b = 0.50$$
$$\rho = 0.1230$$
$$\omega = 0.930$$
$$u = \frac{3}{10}\,\frac{bh}{M-b} = 0.05 \text{ in.}$$
$$i = 1\tfrac{1}{2}(h+u) = 0.825 \text{ in.}$$
$$C = \frac{Sb^3\rho}{M} = 3844 \text{ lb.}$$
$$\Delta = \frac{S\pi}{G}\cdot\frac{M^2}{b\omega} = 0.676 \text{ in.}$$
$$p = \Delta + h + u = 1.226 \text{ in.}$$
$$n = \frac{H-i}{p} = 7.48$$
$$E = (n+1\tfrac{1}{2})(h+u) = 4.94 \text{ in.}$$
$$L = H - \frac{P}{C}(H-E) = 6.71 \text{ in.}$$
$$B = \pi M(n+1\tfrac{1}{2}) = 113 \text{ in.}$$

The last three quantities are those required.

Example 2

31 To design a rectangular-bar spring of 4 in. maximum outside diameter, $2\frac{1}{2}$ in. minimum inside diameter, to carry 1000 lb. at a height of 6 in., with a maximum reserve movement (from loaded height to solid height). Allowable fiber stress 140,000 lb. per sq. in., as the normal load is nearly constant. The torsional modulus will be low on account of the high stress, probably about 9,500,000.

32 Allowing a total of $\frac{1}{8}$-in. clearance on the outside and inside diameters for manufacturing variations, it will be seen that the width of the section can be $\frac{7}{8}$ in. It can be shown that to meet the requirement of maximum reserve movement, the thickness of bar should be such that the capacity of the spring is twice the working load, or in this case 2000 lb. Then from the capacity formula we have

$$\rho = \frac{CM}{Sb^3} = \frac{2000 \cdot 3\frac{1}{2}}{140{,}000 \cdot (\frac{7}{8})^3} = 0.0666$$

From the table

$$k = 0.355 \qquad \text{and} \qquad h = kb = 0.311 \text{ in.}$$

Preferably the nearest commercial size, or $\frac{7}{8}$ in. by $\frac{5}{16}$ in., should be selected. Recalculating,

$$k = 0.357$$
$$\rho = 0.0672$$
$$\omega = 0.700$$
$$C = 2016 \text{ lb.}$$
$$\Delta = 0.738 \text{ in.}$$
$$h+u = 0.350 \text{ in.}$$
$$i = 0.525 \text{ in.}$$
$$p = 1.088 \text{ in.}$$

Deflection per coil under 1000-lb. load $= \Delta\,\frac{P}{C} = 0.366$ in.

Pitch under 1000-lb. load $= p - 0.366 = 0.722$ in.

$$\text{Number of active coils} = \frac{L-i}{0.722} = 7.58$$

$$H = np+i = 8.78 \text{ in.}$$
$$E = n(h+u)+i = 3.18 \text{ in.}$$

and the bar length can be determined as in Example 1. It would be advisable to check the correctness of the calculations through the formula

$$\frac{P}{C} = \frac{\text{movement under load}}{\text{total movement}} = \frac{H-L}{H-E}$$

Example 3

33 A spring of square steel, 12 in. free length, 6 in. maximum outside diameter, no limit on inside diameter, to have a scale deflection of 8500 lb. per in. (i.e., a movement of 1 in. for each 8500-lb. increment of load). While the service is not severe, and the maximum loads are moderate, it is desired that the scale deflection be maintained as constant as possible indefinitely. Hence the very moderate stress of 100,000 lb. per sq. in. will be adopted as a maximum. In that case the torsional modulus will run about 11,000,000.

34 This problem must be solved by cut and try. It is possible to construct an equation for the required size of bar, in terms of the conditions given; but it is a complex one involving fifth powers, therefore is not resolvable, and is not even convenient for use by the cut-and-try method. Assuming various sizes, and working through the steps described in the foregoing problems, we find that a 1¼-in. square bar will make a spring of the following characteristics:

$C = 17{,}110$ lb.
Total movement = 2.14 in.
Scale deflection = 8000 lb.

35 This scale deflection being somewhat greater than the desired figure, the outside diameter should be decreased to make the spring stiffer. By making the outside diameter 5⅞ in., or $M = 4\frac{3}{8}$ in., we get

$C = 17{,}580$ lb.
$\Delta = 0.362$ in.
$h+u = 1.389$ in.
$i = 2.084$ in.
$H-i = 9.916$ in.
$p = 1.751$ in.
$n = 5.66$
$E = 9.95$ in.
Total movement = 2.05 in.
Scale deflection = 8580 lb. per in.

which is sufficiently close to the required figure.

36 Had the scale deflection found on trial been less than the specification instead of more, the remedy would have been to increase the solid height of the spring somewhat; or the next smaller commercial size of bar might be used, with a decreased outside diameter.

Example 4

37 A safety-valve spring of square steel, 5 in. free length, to operate over a valve seat of 2-in. effective diameter with valve closed, and 2½ in. diameter with valve open. The lift of the valve is to be at least ¾ in. Popping pressure 200 lb. per sq. in. The spring should have at least ⅜ in. reserve movement below its length with valve open, and the

outside diameter is to be held, if possible, within $2\frac{1}{2}$ in. Required the size of bar, and the length with valve closed. The fiber stress should not exceed 90,000 lb. per sq. in., and the torsional modulus can be taken at 11,500,000.

38 First, a word of comment on a special practice in the design and manufacture of safety-valve springs. While helical-spring theory assumes the external load, and the spring reaction, as acting along the axis of the helix, it is obvious that this reaction must really be delivered from the end of the active coils. This unsymmetrical pressure, when applied to a floating or semi-floating member such as a safety-valve seat, is apt to cause some very undesirable results. By increasing the dead length of the spring somewhat, it is possible to reduce this lack of symmetry in the spring reaction very materially. Therefore it is customary, in valve-spring practice, to press the dead or bearing coils down tight on the adjacent coils for about one-half a turn back of the tips. Then the number of dead or inactive turns will be about 2, and the dead height $2(h+a)$.

39 Proceeding to the design:

$$\text{Load with valve closed} = \frac{\pi}{4}\ (2)^2 \times 200 = 628 \text{ lb.}$$

$$\text{Load with valve open} = \frac{\pi}{4}\ (2\tfrac{1}{8})^2 \times 200 = 982 \text{ lb.}$$

Then, since the spring is to compress $\frac{3}{8}$ in. between these loads, we can at once write down, from the proportionality of loads and deflections, a theoretical load test for the spring desired, as follows:

Load, lb.	Deflection, in.	Length, in.
0	0	5.000
628	0.665	4.335
982	1.040	3.960
1335 (Capacity)	1.415	3.585 (solid)

40 From this point on, the cut-and-try method must be followed. We then find that the conditions cannot be fully met, with the outside diameter limited to $2\frac{1}{2}$ in. A spring of $\frac{7}{16}$-in. square steel will have more than the required capacity but not sufficient movement, while with $\frac{1}{2}\frac{5}{2}$-in. square steel the reverse conditions obtain. Solving the capacity equation for the mean diameter at which a spring of $\frac{7}{16}$-in. square steel will have just the capacity desired, we find it to be 2.350 in., or an outside diameter of 2.788 in. Then going through the formal calculations we find the corresponding solid height to be 3.66 in., still a little high. But to go to a $\frac{1}{2}\frac{5}{2}$-in. square bar (the next commercial size) would mean a considerable further increase in the outside diameter. By increasing the maximum fiber stress to 92,500 lb. per sq. in., and the outside diameter to 2.851 in., the $\frac{7}{16}$-in. square will just meet the required test.

41 The slight increase in fiber stress would not be objectionable, but the designer must now decide whether to modify his valve design to accommodate the increased outside diameter of spring just arrived at, or to resort to a more special and expensive spring, of rectangular steel. If the latter alternative is chosen, the procedure would be to assume a width of bar section, determine through the capacity equation the corresponding thickness of bar for the capacity required, and then check up on the total movement as before.

42 Here the designer should observe that the wider and thinner the bar for a given capacity, the greater the total movement. Therefore, in the trial calculations, the greatest width consistent with manufacturing limitations should be selected — not more than one-quarter of the

outside diameter — and preferably not more than one-fifth. In the present case, using a fiber stress of 100,000 lb. per sq. in. (which would be quite as conservative as 90,000 in a square bar), and a modulus of 11,000,000, it will be found that a spring of $\frac{1}{8}$-in. by $\frac{11}{32}$-in. steel will give both capacity and movement slightly in excess of the necessary figures, and the requirements of the problem will be met within practical limits.

43 The four examples presented above are typical of the diverse problems which the spring designer meets in practice. As in all other branches of machine design, and perhaps to a greater extent than in most, he must arrange and rearrange his working formulas to meet the needs of the particular problem in hand. In the occasional instance where a square or rectangular is indicated as the proper solution, it is the author's hope that the methods and examples set forth in the foregoing discussion may be of some assistance.

DISCUSSION

W. L. DeBaufre.[1] The author points out the lack of formulas for calculating any except springs of bar steel. The writer desires to point out his contribution to this subject which appeared in the *Journal* of the American Society of Naval Engineers in May, 1917, vol. 29, pages 268 to 299, inclusive.

In testing springs for safety valves at the Naval Engineering Experiment Station at Annapolis, Md., the writer encountered the same difficulty mentioned by the author. He also referred to the work of St. Venant. In deriving formulas for safety-valve springs, it was found that four factors were involved in the strength and rigidity of the spring, namely, the direct tension or compression to which the bar is subjected by the load, the direct shear due to this load, the bending due to the load, and the torsion due to the load. In ordinary spring calculations the last-mentioned only is involved. St. Venant includes also the effect of bending, but the writer's analysis shows that the effect of direct shear was important while that due to bending could be neglected.

It is understood that this article has been used by a large spring manufacturer and found to agree very closely with practice, so that springs of irregular cross-section can now be calculated with assurance of the spring as made corresponding to the calculated results, while previously it was simply a matter of trial and error to find a spring which would meet given specifications.

S. Timoshenko.[2] The intention of the author was to give spring designers some necessary information on the torsion of bars of rectangular cross-section. This problem on torsion was studied in much detail very long ago, and we think it was unnecessary to repeat the calculations again. The complete solution of the problem was obtained by St. Venant in 1855 and published in Memoires des Savants Étrangers, vol. 14, pp. 233-500. The most important

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[2] Research Engineer, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa. Mem. A.S.M.E.

parts of this memoir can be found in the well-known book A History of the Theory of Elasticity, by T. Todhunter and K. Pearson, vol. 2, part 1. The paper of St. Venant contains not only the general solution in the form of an infinite series, but also approximate formulas and numerical tables which are very useful for engineers.

The approximate formula for maximum shearing stress is of the form

$$S = \frac{M(3b+1.8h)}{b^2h^2}$$

where S = maximum shearing stress
b = width of section
h = thickness of section
M = twisting moment.

This formula, which can be found in the book Applied Mechanics, by C. E. Fuller and W. A. Johnston, page 383, gives results with an error of less than four per cent in comparison with the exact value, and is always accurate enough for design.

In addition to his table, St. Venant calculated a numerical table for figuring stress and twist in a bar of rectangular section. The final formulas for these quantities are given in the same form as in the paper, and the numerical factors entering in these formulas are calculated for ratios b/h from $b/h = 1$ to $b/h = 100$, with small intervals so that the intermediate values of these factors can be obtained by simple interpolation.

It must be noted that the statement in Par. 5 is not exact. The approximate formulas mentioned are not based on the assumption that the section of the bar retains its original shape (on such assumption the maximum stress takes place at the corner of the section), but on a certain assumption about the stress distribution over the section, such that conditions at the outline of the section are satisfied.

The writer cannot agree with the specimen calculation given in Par. 18. In evaluating the summation of the series the table of hyperbolic functions must be used. Such a table will make it unnecessary to calculate such functions as $\cosh x$ and $\tanh x$.

THE AUTHOR. Possibly the purpose of this paper has been misunderstood. No claim is made for originality with regard to the basic principles involved. Several authorities who mention the St. Venant formulas give examples of their use, as applied to spring design and other problems.

But the author has never found any article that embodied definite and complete instructions for the guidance of the average spring designer, who generally has neither the time nor the inclination to carry general formulas through the mathematical work required to adapt them to his particular problem. The author therefore attempts to supply just such instructions. For the benefit of the engineer who is interested in the "Why" as well as the "How," the methods by which the working formulas are derived are included.

No. 1979

AN OUTLINE FOR THE APPLICATION OF FATIGUE AND ELASTIC RESULTS TO METAL-SPRING DESIGN

By T. McLean Jasper,[1] Urbana, Ill.
Non-Member

This paper deals particularly with factors of design of metal springs used for shock-absorbing purposes and for recuperating machinery. The problem of the design of springs, the author states, is divided into two parts: (1) *the static elastic and fatigue properties of the material to be used in their construction; and* (2) *the shape of the springs, together with the distribution of the stresses developed in their use for a given deformation. In the discussion of these two points the author gives results of experimental work carried out at the Engineering Experiment Station of the University of Illinois.*

SPRINGS are used as mechanisms to decrease shock, as a part of recuperating machinery, as force-measuring or load-weighing devices, as mechanisms for electrical vibrators, telephone and wireless sending and receiving instruments, and as a storage of energy or as a secondary source of energy in balancing mechanisms.

2 The most extensive uses to which springs are put are for shock-absorbing and recuperating purposes. In the use of springs for such purposes the question of human safety and comfort becomes most important and for this reason the principles underlying the use of metals for such springs need to be most carefully investigated. Springs intended to be used as secondary sources of energy, as weighing devices or as vibrator mechanisms, while important in themselves, can be constructed with a lesser knowledge as to stress conditions or history of the material without seriously impairing their usefulness or risking disaster, should failure occur. Shock-absorbing and recuperating springs are made almost entirely from wrought steel. Almost all the mechanisms on which human safety and comfort depend have this class of

[1] Special Research Associate Professor of Engineering Materials, University of Illinois.

Contributed by the Special Research Committee on Metal Springs and presented at the Annual Meeting, New York, November 30 to December 4, 1925, of The American Society of Mechanical Engineers.

spring associated with them, and for this reason the governing factors in this particular part of the field of spring design will be taken up in this paper.

3 The problems of spring design fall into two general classes: (*a*) the class in which the question of the static, elastic, and fatigue properties of the material used in their construction is considered; and (*b*) the class in which the question of the shape of such springs, together with the distribution of the stresses developed in their use for a given deformation, has to be considered.

4 The elastic properties of the materials used in springs, such as the modulus of elasticity and modulus of rigidity, or shearing modulus, properly combined with the appropriate section modulus

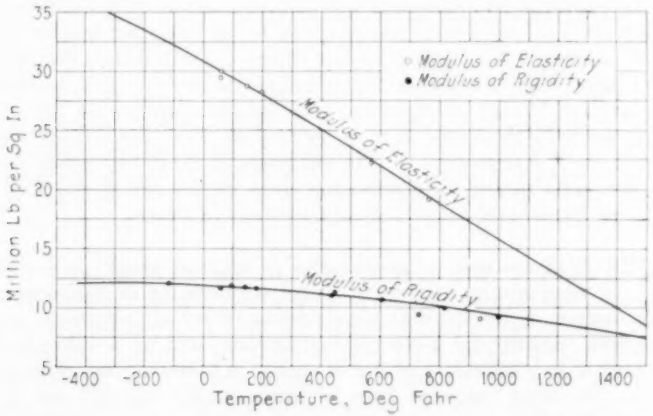


FIG. 1 VARIATION OF THE MODULUS OF ELASTICITY AND MODULUS OF RIGIDITY WITH TEMPERATURE FOR STEEL

(Heavy lines represent theoretical values; plotted points represent experimental values.)

of the shape, determine the stiffness of a spring or its load-deflection rate. The static and fatigue properties of materials used in springs, such as the elastic limit or endurance limit, properly combined with an appropriate working factor, determine the maximum stress that may be developed in any material used for springs. The maximum usefulness of a spring is found when the above factors are correctly combined to calculate the maximum energy that can be absorbed by the spring within its working range.[1]

5 The problem of spring design, therefore, is, first, that of material; second, that of shape. The best materials for springs used for load-carrying or recuperating purposes are those capable

[1] See formulas developed by J. K. Wood in A Code of Design for Mechanical Springs, *Mechanical Engineering*, Sept., 1925, p. 713.

of developing a large range of stress without overstraining in such a manner as to produce permanent and progressive deformation. In other words, materials are desirable which are capable of absorbing large amounts of energy per unit of volume within their elastic working range. If springs are to be used where deformations are repeated many times, this working range of stress should be below the endurance limit of the material. For wrought steel suitable for springs this endurance limit is usually considerably below the elastic limit of the material.

6 The question of the design of springs to be used at elevated or depressed temperatures involves knowledge of the values of the modulus of rigidity and modulus of elasticity at such temperatures. For wrought steel this information is available within a large range of working temperatures. For non-ferrous metals such information is somewhat scanty, and much work should be done on the elastic constants for these metals. In general, it is known that as the working temperatures are increased, the moduli values decrease progressively. Fig. 1 shows the theoretical and experimental values obtained for steel at various working temperatures. These curves are reproduced from published data by the author [1] and represent theoretical curves based on the Kinetic Theory of Solids, by Sutherland,[2] and on known experimental values of the moduli at various temperatures. It would seem that for temperatures up to 800 deg. fahr., the needed elastic properties are fairly well established both theoretically and experimentally.

7 The question of the relation between the fatigue and static strength of steel and the temperature at which it is to be used is important. In general, for spring steel, it is known that as the temperature is elevated the fatigue and static-strength values decrease in a progressive manner. Fig. 2 is a reproduction of a diagram from a paper by the author,[3] showing the effect of temperature on the static and fatigue properties of a 1.02 per cent quenched carbon steel. The speed at which this metal was tested in repeated stressing was 1500 cycles per minute. It is suspected that at certain elevated temperatures, as the speed of testing is decreased for any particular steel, the endurance limits will decrease also.

8 The question of using steel at ordinary temperatures for springs has associated with it the mechanical treatment, such as rolling, wire drawing, and coiling, and the heat treatment which the material has received up to the time the finished product is ready to be introduced into a mechanism. Fig. 3 shows the effect

[1] T. M. Jasper, The Value of the Energy Relation in the Testing of Ferrous Metals at Varying Ranges of Stress and at Intermediate and High Temperatures, *Phil. Mag.*, vol. 46, Oct. 1923, p. 609.

[2] Wm. Sutherland, Kinetic Theory of Solids, *Phil. Mag.*, 1891.

[3] T. McLean Jasper, Typical Static and Fatigue Properties of Steel at Elevated Temperatures. A.S.T.M. Proceedings, 1925.

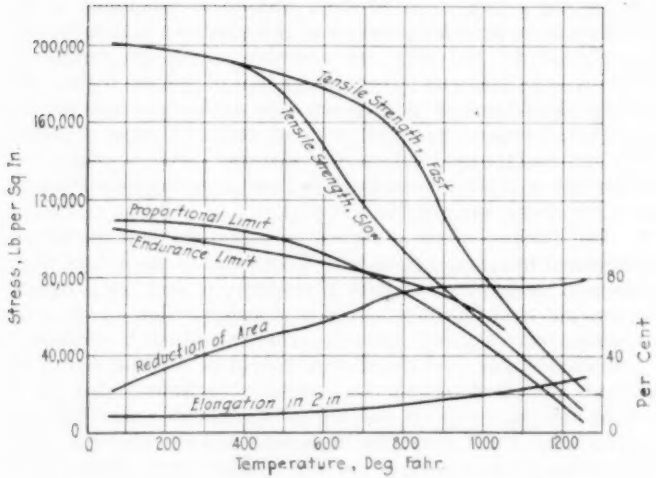


FIG. 2 EXPERIMENTAL RESULTS OF TESTS OF A 1.02 PER CENT CARBON STEEL QUENCHED AND TESTED AT VARIOUS TEMPERATURES

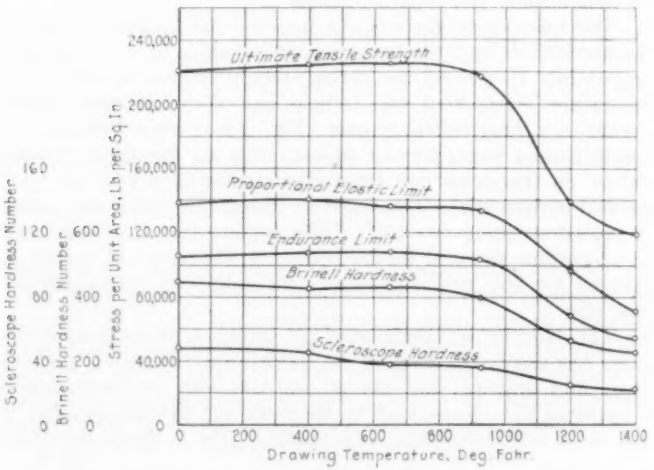


FIG. 3 CURVES SHOWING CERTAIN PHYSICAL PROPERTIES OF 1.20 PER CENT CARBON STEEL QUENCHED FROM ABOVE THE CRITICAL POINT AND DRAWN AS INDICATED

of heat treatment on the static and fatigue properties of 1.20 per cent carbon steel when it is tested at ordinary working temperatures. It shows the effect of quenching from above the critical

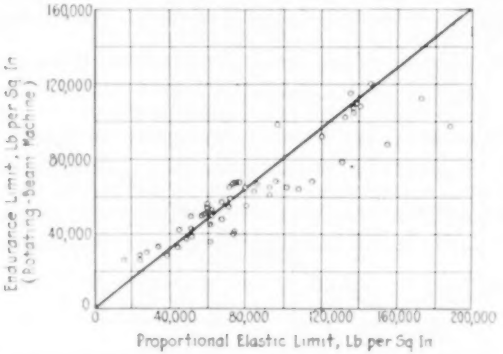


FIG. 4 CORRELATION OF ENDURANCE LIMIT WITH PROPORTIONAL ELASTIC LIMIT (TENSION)

point combined with various drawing treatments. In general, it may be said that drawing a carbon steel up to 800 deg. fahr. has very little effect in decreasing the physical strength properties

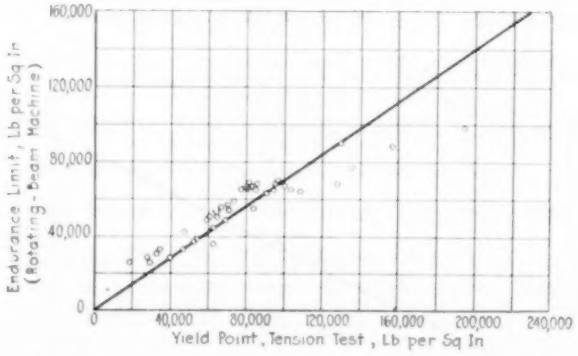


FIG. 5 CORRELATION OF ENDURANCE LIMIT WITH YIELD POINT (TENSION)

shown in Fig. 3, but for a nickel alloy steel, the point at which such properties begin to decrease rapidly is somewhere in the neighborhood of a draw of 400 deg. fahr.[1]

[1] Bulletin 136, Engineering Experiment Station, University of Illinois.

9 When springs are to be designed of steel where stresses are to be repeated many times, the allowable working stress should depend somewhat on the type of work the springs are to do and on the relative human hazard introduced by the mechanism in which they are to be used. It would seem that the endurance-limit

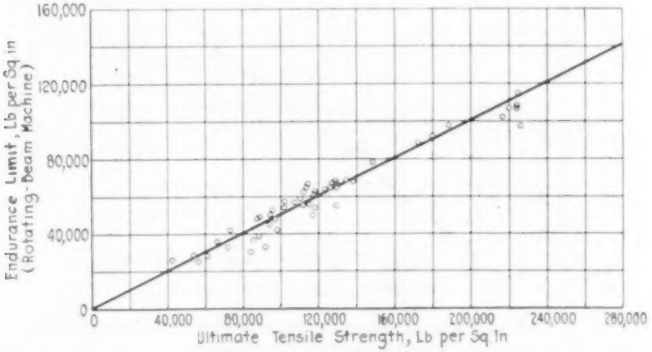


FIG. 6 CORRELATION OF ENDURANCE LIMIT WITH ULTIMATE STRENGTH (TENSION)

stress is the strength property to use with an appropriate method of determining a factor of safety, rather than the elastic limit of the material. Figs. 4, 5, 6, and 7 show the correlation between the endurance limits and static properties of several steels.

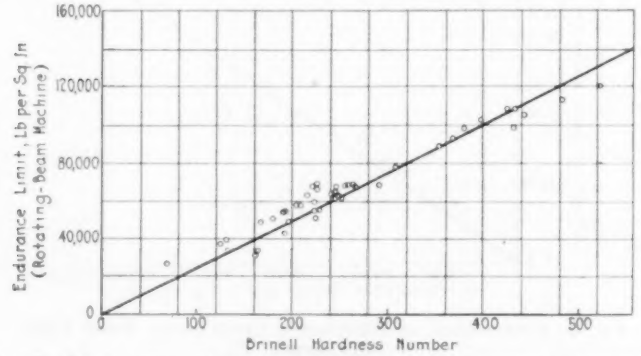


FIG. 7 CORRELATION OF ENDURANCE LIMIT WITH BRINELL HARDNESS

10 The results given in Figs. 3 to 7 are taken from investigations carried out by the Engineering Experiment Station of the University of Illinois.[1] Fatigue results consume much time and require

[1] H. F. Moore and T. M. Jasper. An Investigation of the Fatigue of Metals. Bull. 136, University of Illinois Engineering Experiment Station, Urbana, Illinois.

a considerable amount of special apparatus. Since the correlation is fairly close between certain simple static-test results and endurance-test results, it is suggested that the static properties which give closest correlation to the endurance limits be used as a basis of allowable stress in spring design. Such correlation of results, however, can only be applied to sound steel. The most appropriate static properties are shown in Figs. 6 and 7.

11 The type of stresses developed in springs is that of flexure or of torsion, or of some combination of flexure and torsion.

12 The ratio of endurance limits found between complete reversal of stress in torsion and complete reversal of stress in flexure for 19 different steels and heat treatments is shown in Fig. 8. Fig. 8 shows that the average ratio between endurance limits in torsion and in flexure is in the neighborhood of 0.53. Fig. 8 is

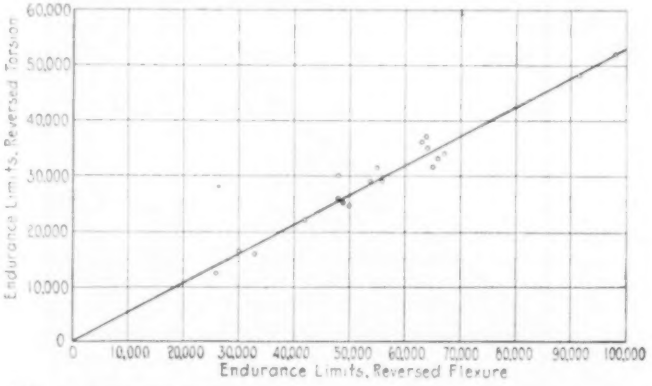


FIG. 8 RELATION BETWEEN ENDURANCE LIMITS IN TORSION AND IN FLEXURE FOR 19 STEELS

(Tested in Fatigue of Metals Laboratory, University of Illinois.)

obtained for endurance limits of steel when the stress is completely reversed.

13 In the use of springs, in general, stresses vary from approximately zero to a maximum when any small portion of stressed material is considered, and when the metal is repeatedly stressed under conditions of non-reversal of stress, it is found that for steel the endurance limits in flexure are about 71 per cent of the ultimate tensile strength. It will be shown later that, as a basis of design for steel springs, an allowable unit stress in flexure of 50 per cent of the ultimate strength in tension will give a safety factor of 2 within the endurance-limit range of the metal, if the maximum energy absorbed per unit volume is used. If the allowable unit stress in torsion is 25 per cent of the ultimate strength in tension, approximately the same factors of safety will prevail for springs in torsion.

14 The above numerical values are used as examples only and are not intended as suggestive of the correct factors of safety to be used in spring design. These values apply to spring steel and cannot be assumed for low carbon steels.

15 The factor of safety of 2 based on the energy relation within the endurance range for spring steel, referred to above, is found in the following manner. The amount of work that can be done by a spring varies with the energy that can be absorbed by it. See Fig. 9.

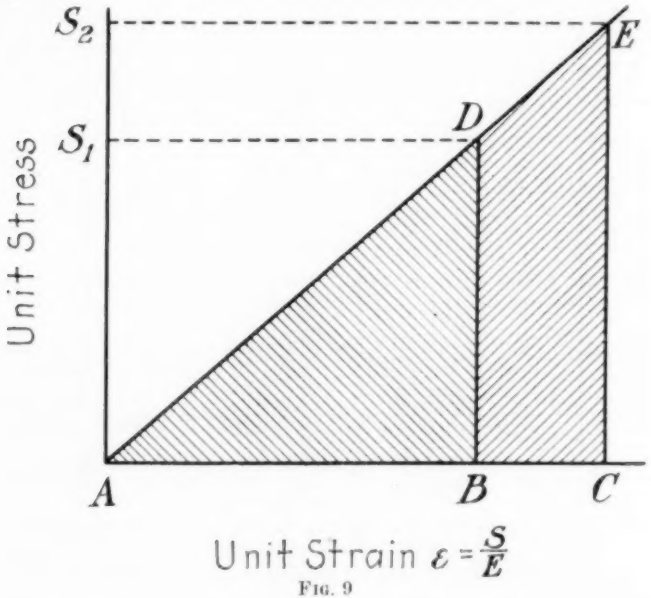


FIG. 9

16 The amount of energy absorbed and given up by a metal when subjected to a cycle of stress varying from zero to a maximum, is shown in Fig. 9 by the area ABD, if S_1 is the maximum value of the stress developed. In spring steel the endurance limit for the condition when the stress is not reversed is found to be approximately 71 per cent of the ultimate strength. The maximum value of the unit energy absorbed in a spring (see Fig. 9) when S_1 is the maximum stress developed, is equal to $S_1^2/2E$, where E is the modulus of elasticity or rigidity, as the case may be. When 50 per cent of the ultimate strength is used, this becomes $0.50^2C/2E$ or $0.25C/2E$, where C is some constant which is in this case the square of the ultimate-strength value. When 71 per cent of the ultimate strength is used, the formula reduces to $0.71^2C/2E$ or $0.50C/2E$. In this manner, when the energy relations are con-

sidered, the factor of safety within the endurance range of the metal is equal to $0.50 \div 0.25$ or 2. The factor of safety in torsion, if 25 per cent of the ultimate strength in tension is used as a working stress, will be in excess of the above value.

17 The author believes, therefore, that if the factors of safety for steel springs are based entirely on the relative energy absorbed and on the endurance limits, this is a more representative method than if the ratio of a limiting stress is used as a basis. The actual factor used might be made more than 2, based on the above calculations for steel springs if occasional overstressing is expected. Overstressing, however, can be avoided by the limiting of the total deformation of springs beyond a certain point in the direction of the deflection. The above discussion applies to steel springs only. Steel springs are used almost entirely when human life and comfort depend on such mechanisms.

18 For springs made of non-ferrous metals, a large amount of information on the properties of such metals must be obtained before a procedure covering the factors to use in design can be outlined in as satisfactory a manner as for steel springs.

DISCUSSION

H. A. F. Campbell.[1] Fig. 3 is an interesting set of curves because, as stated, it deals with one per cent carbon manganese-silicon spring steel. This is the regular grade of carbon steel used by railroads and locomotive builders for driving, engine-truck, and tender-truck springs, and electric-truck and car springs. The ultimate tensile strength, that is, the breaking point, per square inch within the wide range of drawing temperatures from 0 deg. to 800 deg. fahr., is given as 220,000 lb. per sq. in. The endurance limit, when there is no reversal of stress, but an indefinite number of applications of this stress, is given as 71 per cent of the ultimate strength, that is, as 156,000 lb. per sq. in.

Par. 13 states, "It will be shown later that, as a basis of design for steel springs, an allowable unit stress in flexure of 50 per cent of the ultimate strength in tension will give a safety factor of 2 within the endurance limit range of the metal, if the maximum energy absorbed per unit volume is used." Fifty per cent of the 220,000 lb. per sq. in. is 110,000 lb. per sq. in., which the author states can be used as the allowable working unit stress in flexure for plate-spring design. From a practical standpoint, the writer does not believe that a plate spring for a locomotive driving, engine-truck, or tender-truck, or an electric-truck spring, when made of 1 per cent carbon manganese steel, would last long in actual service, with any such working fiber stress.

[1] Assistant to Consulting Vice-President, The Baldwin Locomotive Works, Philadelphia, Pa.

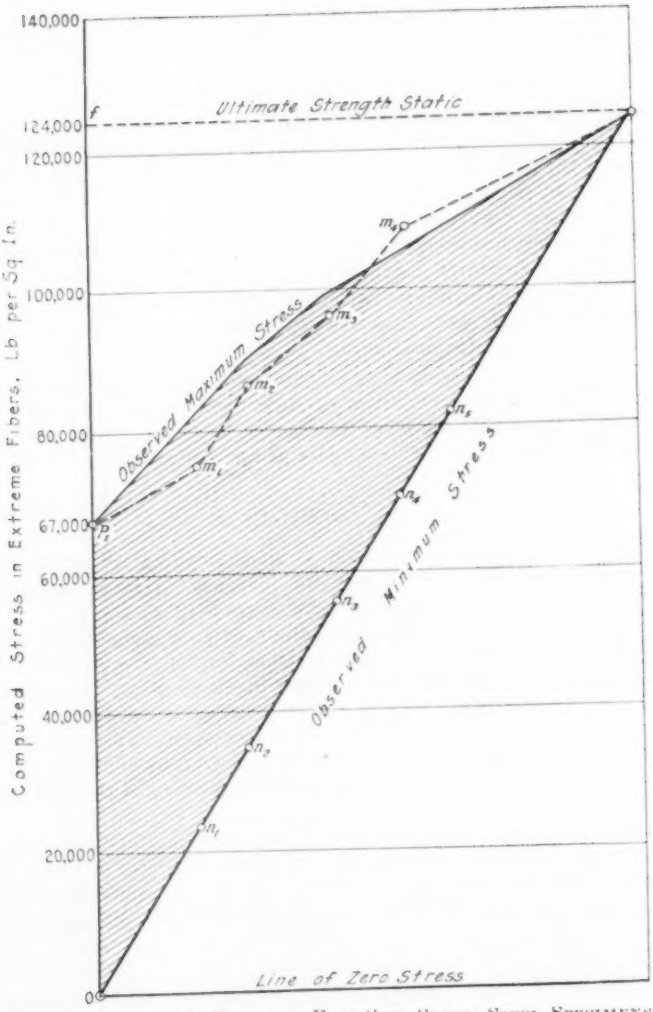


FIG. 10 WOHLER'S TESTS ON FULL-SIZE SPRING-STEEL SPECIMENS

The shaded area is the field in which the material may be worked indefinitely.

$$m = p_1 + \frac{n}{m}(f - p_1)$$

where
m = maximum stress
p_1 = repetition limit when $n = 0$
n = minimum stress
f = ultimate static strength

Factors:
Static load limit = f = ultimate strength
Repeated load limit = p_1 = one-half ultimate strength
Reversed load limit = p_2 = one-third ultimate strength

If we consider full-sized spring-steel plates that vary from $\frac{1}{4}$ in. to $\frac{5}{8}$ in. in thickness, when made of the best quality of carbon manganese-silicon steel, carefully treated, the ultimate tensile strength will be actually about 125,000 lb. per sq. in., and the elastic limit about 95,000 lb. per sq. in. (not over 100,000 per sq. in.).

Driving springs, engine-truck and tender springs, electric-truck springs, and car springs have to withstand a certain static load, plus an increase of this static load of 30 to 50 per cent, and this change of load is repeated in the one direction thousands of times. The working fiber stress under the actual known static loads should not be over 60,000 lb. per sq. in. A 50 per cent increase makes this 90,000 lb. per sq. in. The steel, then, will be working in the range between 60,000 lb. to 90,000 lb. per sq. in. Springs, if cor-

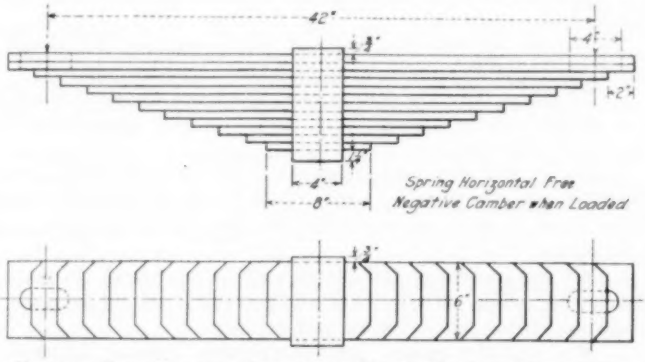


FIG. 11 SEMI-ELLIPTIC SPRING FOR HEAVY TYPE OF LOCOMOTIVE

12 plates, 6 in. x $\frac{5}{8}$ in.; static working load, 26,700 lb.; fiber stress (working), 60,000 lb. per sq. in.

rectly designed, when working within this range, and when made of the best-quality material, carbon manganese-silicon steel, accurately heat treated, should give a good length of service. It must be remembered that these springs, besides working repeatedly in the range between 60,000 to 90,000 lb. per sq. in., must also withstand occasional heavy, suddenly applied shock loads that are actually hammer blows.

Fig. 10 reproduces the well-known tests on full-sized spring-steel specimens made by Wöhler. These tests most clearly show the zone areas within which this spring steel can work.

The resilience of any spring design is usually given too little consideration. Everything possible must be done to obtain as much resilience as possible. The best results will be obtained by using a long spring, say, from 38 to 44 inches, with a few thick plates rather than a greater number of thin plates. These fewer and thicker plates should be most carefully arranged as to length

of plates, number of full plates at end, the length of short plates, and the form of the ends of the plates. Fig. 11 shows a modern semi-elliptic spring that is used under the heaviest locomotive of its type today. This spring can be considered an example of careful design because —

a The length of spring from center to center is 42 in.

b A few thick plates are used.

c The number of full plates at the end, the length of the short plate, the arrangement of the slot and the third long plate, and the tapering of the ends of the plates, are very good design.

d The fiber stress under the actual static load is 60,000 lb. per sq. in.

e This spring is designed to be built straight, free set, and to stand, when under the locomotive, with a negative camber. From the viewpoint of practical manufacturing, this spring also is a good one to make and heat-treat.

The writer believes that some of the spring troubles of today under our heavy locomotives and cars are caused because —

a We allow only the same space, often not as much space as was allowed under lighter units, and into this same space we try to put a spring.

b We try to carry these greater loads on the same 36-in. center-to-center spring with many $\frac{3}{8}$-in. or 7/16-in. plates, when we should use a 42-in. or 44-in. spring with $\frac{1}{2}$-in. or 9/16-in. plates, or even $\frac{5}{8}$-in. plates.

c We use too many full plates and do everything to make a stiff spring, while actually everything possible should be done to make the spring more flexible and resilient.

d We call the elastic limit somewhere around 130,000 to 150,000 lb. per sq. in., when actually it is only 90,000 to 100,000 lb. per sq. in.

e We heat-treat the steel in a too inaccurate and inexact way.

R. Eksergian.[1] The author opens up an important subject on the design of springs as to the limiting fiber stress to be used and as to the fatigue limit being a basic criterion. The data submitted in his paper are worthy of careful study. In the first place, Figs. 6 and 7 indicate that the endurance limit is directly proportional to the ultimate strength and Brinell hardness, while Figs. 4 and 5 indicate some correlation between a direct proportionality between endurance limit and elastic limit and yield point.

The equation of endurance limit with respect to ultimate strength, Fig. 6, is approximately $f_d = 0.5f_v$ where f_d = endurance limit, lb. per sq. in., and f_v = ultimate strength, lb. per sq. in., for complete reversal of stress. In Fig. 4, $f_d = 0.8f_e$ where f_e = elastic limit, lb. per sq. in. Therefore, $f_e = \frac{5}{8}f_v = 0.625f_v$. Thus we have approximately a constant ratio of elastic limit to ultimate strength.

[1] Engineer, The Baldwin Locomotive Works, Philadelphia, Pa. Mem. A.S.M.E.

This, of course, is inconsistent with the range of test specimens used, i.e., test specimens from 80,000 to 200,000 lb. per sq. in. ultimate strength, which even with heat treatment are not likely to have a constant ratio of elastic limit to ultimate strength. So again the question arises whether the endurance limit is only a function of the ultimate strength and Brinell hardness, or is a complicated function of ultimate strength, elastic limit and ductility.

It is felt that test specimens ordinarily used for endurance tests are by no means indicative of the real properties of this material as a machine member. Thus in a spring leaf there may be a considerable difference in Brinell hardness on the inner and outer surfaces, while again as the specimen is increased in size we have a redistribution of stress concentration, initial stresses, etc., and a possible change in ductility and grain structure across the section.

If the specimens in Fig. 8 are subjected to pure bending and torsion, respectively, the maximum principal stress with bending would occur at the outer fiber. Therefore, we have a secondary maximum shear stress on a plane at 45 deg. to the axis at the outer fiber, of value equal to one-half the tension at the outer fiber. This, of course, is comparable with the simple shear stress due to torsion. The results shown in this curve tend to prove that the incipient fractures due to fatigue start on planes of maximum shear. Otherwise, we should expect no direct relation between direct tension in bending and the secondary shear of bending which corresponds to the torsional shear. Moreover, this appears to hold for brittle materials, provided that high-carbon specimens are included in the tests. Thus for a specimen subjected to only one principal stress, the maximum shear is approximately 0.5 of the maximum direct stress, and therefore we should expect the endurance limit, when measured as a direct stress or as a shear stress, to have an invariable proportional relation of approximately 2 to 1.

The writer entirely agrees with the author's recommendation as to the use of the energy cycle for measuring and giving proper values for endurance limits. Thus, assuming a complete reversal endurance limit at 0.5 of the ultimate, the total change in energy is proportional to $[0.5^2 + 0.5^2 = 0.5]$. Therefore, for a simple oscillating stress ranging from 0 to its maximum, the endurance limit would be $\sqrt{0.5} = 0.71$ as given by the author. It is thus seen that since the endurance stress comparison for estimating the real factor of safety is based rather on the square than on a direct comparison of stresses, small oscillations are quickly damped down as to their importance in causing fatigue. Moreover, if fatigue failure can be directly associated with the difference in elastic energy in a cycle oscillation, we can arrive at some idea of the importance of a few oscillations of large amplitude superimposed on a set of constantly occurring low-amplitude oscillations.

Though the author has already submitted valuable data, it would seem for immediate use in spring design that direct fatigue tests should be made on a variety of springs and, moreover, that a study should be made of the relative importance of a few oscillations of large amplitude occurring with different values of mean amplitude.

J. M. LESSELLS.[1] In Par. 9 it appears that the endurance limit rather than the elastic limit is advocated as a basis of design. With this the writer entirely agrees. It must be observed, however, that the majority of springs work under conditions in which the stress does not change sign and in some cases only fluctuates within narrow limits. The basis of design should therefore not be the endurance limit for the complete stress reversal as shown in Figs. 3 and 7, but for stress variation corresponding to the working condition. While the indications are that the endurance limit for complete stress reversal may be a good indication of the endurance limit for other stress conditions, the only justification for such acceptance must be the absence, at present, of endurance-limit data for stress variations. This compromise must be noted. When, however, the author proposes to take the ultimate strength as a guide for determining working stresses, his action must be criticized. In spring steels large residual stresses may exist, and from tests made by the writer it has been shown that the endurance limit bears no definite relation to the ultimate strength.

The writer therefore recommends that working stresses be based on the endurance limit corresponding to the condition under which the springs are working, and if these endurance limits be in excess of the elastic limit, then this latter value should be the basis on which working stresses are based.

THE AUTHOR. The author is gratified by the extent and quality of the discussion which has been brought out by his paper. The remarks that he is offering, therefore, are merely for the purpose of qualifying certain points which have been emphasized by those who have discussed the paper.

Referring to Mr. Campbell's discussion, the point should be brought out that in the design of a spring, when any percentage of ultimate strength is used, this value of the stress should be below the elastic limit of the metal. Discussing the question of factors of safety, the author states in Par. 14 that "the above numerical values are used as examples only and are not intended as suggestive of the correct factors of safety to be used in spring designs." The question of protecting a spring when an occasional heavy applied working load is encountered, by introducing a stop beyond which a spring may not deflect, is undoubtedly feasible.

[1] Engineer in Charge Mechanics, Research Laboratory, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa. Mem. A.S.M.E.

In Mr. Eksergian's discussion the question of a definite percentage of the ultimate strength is again brought out. This point has been taken up in the preceding paragraph. The answer to the question, "Is the endurance limit only a function of the ultimate strength and Brinell hardness, or is it a complicated function of ultimate strength, elastic limit and ductility?" may be partially indicated by referring to Figs. 4 to 8. It is quite possible that all of the enumerated factors enter into this problem, but a close survey of these curves will indicate which factors really give the best correlation. The question of ductility has been studied in this connection, but no relation between it and the endurance limit was indicated. The author is heartily in sympathy with the idea that direct fatigue tests should be made on a variety of springs. He would suggest, however that, when this is done, it will also be necessary to use formulas that conform to the correct stress analysis rather than the simple formulas most commonly adopted for spring designs.

The author is very glad that Mr. Lessells has pointed out that the maximum stress used in springs must not be beyond the elastic limit. In Par. 13 the author has referred to the fact that in most springs the stress has a zero maximum range or nearly so. The remarks in the paper on zero maximum range are based on experimental evidence produced in the reference given in the footnote on page 736. Mr. Lessells states that "from tests made by the writer it has been shown that the endurance limit (referring to spring steels) bears no definite relation to the ultimate strength." As to this, the author respectfully refers to Fig. 6, in which the values of the ultimate strength indicate the spring steels plotted on this diagram. Had Mr. Lessells said that the elastic limit bears no definite relation to the endurance limit for spring steels, the author would have agreed with him entirely.

No. 1980

BOLTS FOR USE IN POWER-PLANT CONSTRUCTION

By William P. Wood,[1] Ann Arbor, Mich.

Non-Member.

In this paper the author discusses the use of steel vs. wrought iron and screw stock for bolts and nuts used in power-plant construction. He concludes that the carbon content for bolting steel ranging from about 0.2 *per cent up to* 0.45 *per cent will yield the best combination of tensile strength and ductility in either the heat-treated or non-heat-treated condition. A tentative specification for high-temperature alloy-steel bolting material is appended.*

THE stock for bolts and nuts is just now beginning to receive careful consideration from designing engineers. Hitherto bolts and nuts have been looked upon as one of the connecting links in construction work, but a link of no importance from a physical-property standpoint. The only requirement was that they should possess proper conformance to certain dimensional features.

2 Such a viewpoint is in process of a revolutionary change. Engineers, especially designing engineers for large power companies, are today giving careful scrutiny to the character of the bolts and nuts used in their plants. This paper covers the findings from one such study.

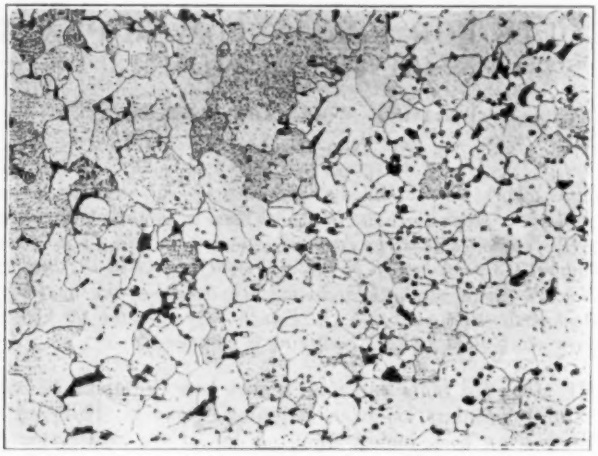
Steel vs. Wrought Iron

3 One of the conditions which became apparent after a little investigation was the indiscriminate use of steel and wrought iron as bolting material. It is unquestioned that there are many instances where this may be done without serious consequences, but is the construction of high-pressure steam lines one of these instances? Granted that the use of wrought iron may be feasible in some instances, it should at least be stipulated that the wrought iron be of as good quality as possible. That this is not always the case may be shown by citing a few examples. The photomicro-

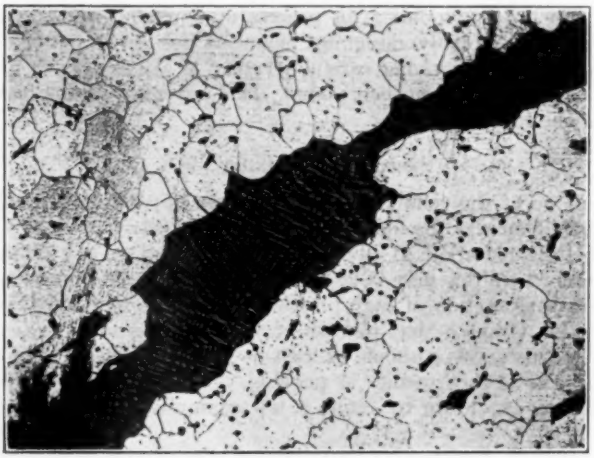
[1] Assistant Professor of Metallurgical Engineering, University of Michigan.

Contributed by the Power Division and presented at the Annual Meeting, New York, November 30 to December 4, 1925, of The American Society of Mechanical Engineers.

graphs shown in Fig. 1 represent sections from two wrought-iron steam-line bolts. Two things stand out in these four photomicrographs: first, the presence of some enormous slag inclusions (the black portions), and second, the extreme non-homogeneity of the



No. 1 Cross section of bolt No. 1. Left-hand portion fairly free from slag; right-hand portion carries much slag.



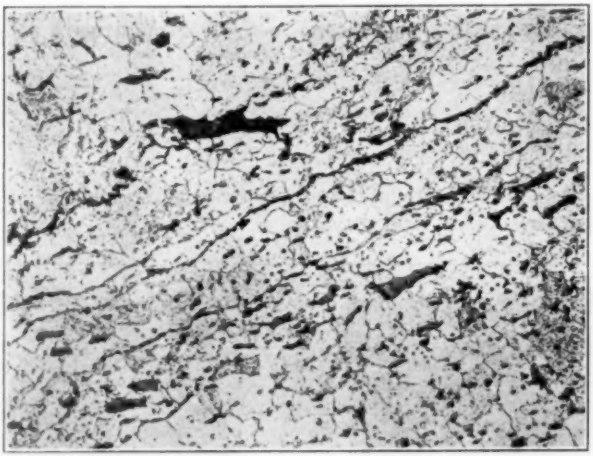
No. 2 Longitudinal section of bolt No. 2. Note the large slag inclusion.

FIG. 1 PHOTOMICROGRAPHS OF SECTIONS OF FOUR WROUGHT-IRON BOLTS

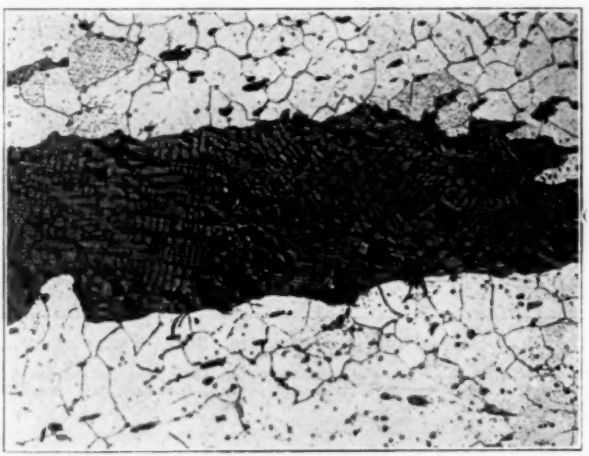
(Continued on following page.)

Etch, 4 per cent picric-nitric acid mixture in alcohol; magnification, ×100.

crystalline portions of the bolts. A good piece of wrought iron should have a structure similar to the left-hand portion of photomicrograph No. 1. Fig. 2 is a slightly enlarged view of the macroscopically etched longitudinal sections of these two bolts. Note the extreme fiber distortion, which has gone so far in some cases



No. 3 Longitudinal section of bolt No. 3. Note the streaked and non-uniform quality of the metal.



No. 4 Longitudinal section of bolt No. 4. Note the large slag inclusion.

FIG. 1 (Continued)

as to actually rupture the surface of the bolt. These two sections were taken immediately under the heads of the bolts. The structure shown here almost suggests that these bolts were "cold headed."

4 The physical properties of these two wrought-iron bolts are given in Table 1. While the ductility of these two bolts is

TABLE 1 PHYSICAL PROPERTIES OF TWO WROUGHT-IRON BOLTS
(Diameter, 0.505 in.; gage length, 2 in.)

Sample	Elastic limit, lb. per sq. in.	Maximum strength, lb. per sq. in.	Elongation, per cent	Reduction of area, per cent
Bolt No. 1.....	35,000	52,500	26.5	46.0
Bolt No. 2.....	41,500	58,500	18.0	44.5

acceptable, the elastic limit and tensile strength should be at least 15 per cent higher than they are.

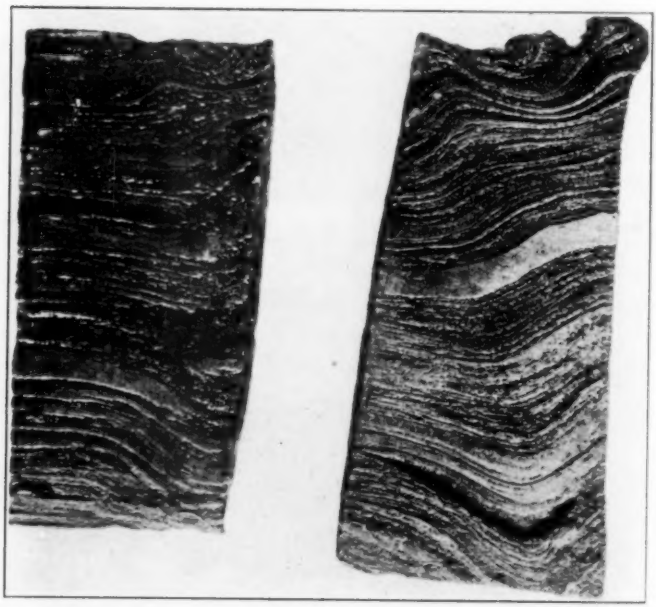


FIG. 2 LONGITUDINAL SECTIONS OF BOLTS NOS. 1 AND 2

(Etched for 1 hr. in 10 per cent sulphuric acid. The bright sections are bands of steely structure, while the dark bands are wrought iron and slag. Magnification, × 3.)

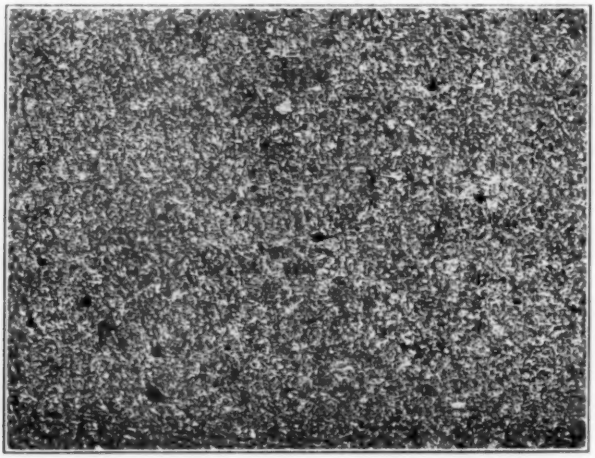
NOTE TO TABLE 2: In carrying out the tensile tests reported in this table, bolts of all sizes were used. It was therefore impossible to prepare standard 0.505-in.-diameter tensile specimens in all cases. Where the diameter was less than 0.505 in., the gage length was determined by the relation $L/\sqrt{(a)} = 4.5$, where L = gage length and a = cross-sectional area of test section.

TABLE 2 PHYSICAL TESTS OF 71 STEAM-LINE BOLTS

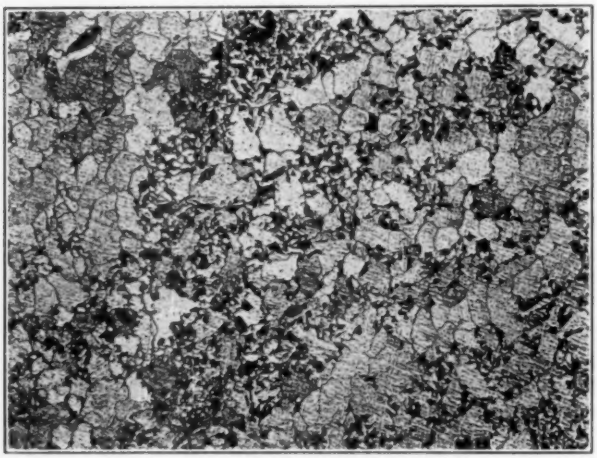
(See note at bottom of preceding page.)

Specimen	Diameter, in.	Elastic limit, lb. per sq. in.	Maximum strength, lb. per sq. in.	Breaking point, lb. per sq. in.	Gage length, in.	Elongation per cent, basis, 2 in.	Reduction of area, per cent	Material
1-H	0.501	31,400	51,700	44,000	2	31.0	46.4	W. I.
1-I	0.501	33,000	51,700	47,500	2	17.5	35.0	W. I.
1-O	0.502	34,800	56,000	50,000	2	16.5	35.0	W. I.
2-A	0.505	33,500	52,500	44,000	2	22.0	48.5	W. I.
2-C	0.501	35,500	53,300	43,600	2	32.5	51.0	W. I.
3-C	0.501	69,500	100,000	62,000	2	26.0	67.0	Steel
4-E	0.502	40,000	56,000	50,000	2	16.5	35.0	W. I.
4-K	0.502	35,000	53,000	47,000	2	24.0	47.0	W. I.
4-N	0.502	38,400	54,500	48,500	2	19.5	37.3	W. I.
5-A	0.376	32,450	52,200	45,400	1.5	23.75	45.0	W. I.
6-A	0.378	43,500	62,500	39,000	1.5	40.0	79.0	Steel
8-A	0.504	34,000	49,500	43,000	2	20.0	41.0	W. I.
9-A	0.501	85,700	122,000	76,000	2	23.0	63.0	Steel
10-A	0.501	46,500	65,500	47,200	2	35.0	64.5	Steel
14-A	0.407	43,850	64,500	44,600	1.62	37.1	58.5	Steel
15-D	0.439	33,100	55,200	39,700	1.75	40.5	63.0	Steel
16-A	0.376	103,500	127,000	84,700	1.5	24.0	51.7	Steel
16-FF	0.376	80,000	140,000	98,200	1.5	20.0	55.2	Steel
16-G	0.376	112,500	136,000	95,000	1.5	20.0	65.0	Steel
16-L	0.376	102,000	121,000	81,000	1.5	22.0	53.2	Steel
16-Q	0.376	110,000	131,500	95,600	1.5	21.3	54.0	Steel
16-Z	0.376	117,000	137,700	93,000	1.5	18.65	65.7	Steel
17-F	0.470	41,500	81,500	58,400	1.87	31.0	60.5	Steel
17-I	0.470	72,500	122,500	72,500	1.87	16.6	63.7	Steel
18-B	0.505	60,000	94,000	69,000	2.0	27.0	60.2	Steel
19-C	0.505	120,000	139,000	105,500	2.0	17.5	49.0	Steel
19-L	0.439	102,500	124,000	87,500	1.75	22.3	55.5	Steel
20-A	0.502	111,500	126,000	74,000	2.0	17.0	65.5	Steel
20-E	0.501	106,700	122,000	69,000	2.0	16.0	67.0	Steel
21-F	0.376	46,900	84,500	61,300	1.5	32.0	59.5	Steel
21-K	0.376	....	95,000	65,000	1.5	19.3	58.6	Steel
21-M	0.376	40,500	69,400	43,200	1.5	36.6	71.6	Steel
22-E	0.376	39,600	70,400	45,000	1.5	37.3	70.0	Steel
22-I	0.376	39,600	69,400	44,100	1.5	37.3	71.2	Steel
22-L	0.376	40,500	68,500	42,300	1.5	38.6	70.8	Steel
23-E	0.219	135,000	148,500	83,800	0.875	20.6	67.0	Steel
24-B	0.221	50,000	65,800	47,300	0.875	19.4	63.2	Steel
25-B	0.256	126,500	140,000	89,300	1.0	19.0	66.0	Steel
29-C	0.250	94,000	116,000	79,500	1.0	20.0	61.0	Steel
30-G	0.346	116,000	131,000	89,000	1.38	29.8	60.0	Steel
30-H	0.315	117,000	135,000	102,500	1.25	21.6	52.2	Steel
30-L	0.314	119,500	142,000	99,000	1.25	20.0	56.0	Steel
32-A	0.313	48,000	56,000	52,000	1.25	12.0	26.0	W. I.
32-A	0.315	37,600	50,500	50,500	1.25	16.0	32.5	W. I.
32-B	0.316	38,400	42,200	60,000	1.25	35.0	66.5	Steel
32-E	0.312	30,000	52,400	44,500	1.25	28.0	47.5	W. I.
32-Q	0.250	40,800	61,200	43,800	1.00	27.0	65.4	W. I.
33-A	0.313	78,000	90,000	67,500	1.25	18.4	49.4	Steel
36-A	0.252	49,000	87,500	67,000	1.00	30.0	42.6	Steel
36-D	0.250	50,000	90,000	65,000	1.00	30.0	58.0	Steel
37-A	0.220	34,200	52,500	52,500	0.875	21.7	35.2	W. I.
37-B	0.251	42,800	55,000	49,000	1.00	21.0	32.6	W. I.
37-D	0.220	40,700	56,500	47,500	0.875	25.7	47.5	W. I.
37-H	0.221	40,700	56,600	47,400	0.875	23.0	47.0	W. I.
39-A	0.439	36,300	53,000	43,500	1.75	31.4	50.0	W. I.
39-FF	0.406	34,000	53,500	45,000	1.62	23.5	43.4	W. I.
39-G	0.439	33,800	54,300	46,400	1.75	25.1	47.6	W. I.
39-I	0.407	45,000	73,000	55,400	1.62	30.8	58.5	Steel
39-H	0.407	31,200	52,300	42,800	1.62	28.4	55.4	W. I.
39-L	0.437	25,200	41,500	35,200	1.96	25.0	56.0	W. I.
41-A	0.252	51,000	143,000	....	1.00	15.0	57.0	Steel
46-B	0.251	38,800	51,000	40,800	1.00	25.0	38.0	W. I.
46-C	0.284	44,300	63,300	47,500	1.125	22.1	9.2	W. I.
47-A	0.284	36,500	53,000	47,500	1.125	31.0	47.5	W. I.
47-B	0.283	36,500	53,000	46,000	1.125	30.0	47.5	W. I.
49-A	0.221	43,500	66,000	47,500	0.875	40.0	66.0	Steel
50-A	0.251	37,000	56,000	47,000	1.00	24.0	47.0	W. I.
50-B	0.251	39,000	55,000	49,000	1.00	28.0	34.8	W. I.
51-A	0.346	44,700	53,200	45,800	1.37	33.5	52.0	W. I.
51-B	0.345	35,500	51,600	45,200	1.37	30.6	53.6	W. I.
51-C	0.347	37,200	53,200	45,700	1.37	25.5	54.3	W. I.

5 Many instances were found where shanks of bolts were made from steel and nuts from wrought iron. One lot of bolts which were investigated were particularly interesting from this standpoint. The bodies of the bolts were made from a hardened 0.4 per cent



No. 1 Longitudinal section of a hardened 0.4 per cent carbon steel bolt.



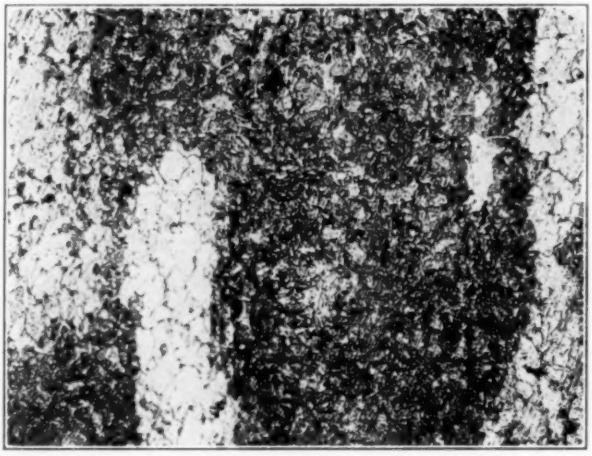
No. 2 Section of nut No. 1 made from bundled scrap.

FIG. 3 PHOTOMICROGRAPHS OF SECTIONS OF A BOLT SHANK AND OF THREE NUTS

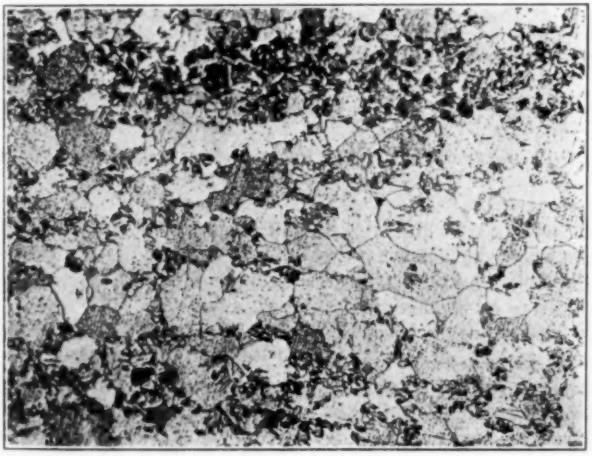
(Continued on following page.)

Etch, 4 per cent picric-nitric acid mixture; magnification, × 100.

carbon steel and possessed unusually high tensile properties coupled with good ductility. The nuts, on the other hand, were made from a poor quality of "bundled scrap." Photomicrograph No. 1 of Fig. 3 represents the structure of the shank of steel bolt No. 3. The structure is sorbitic, which means that it is fairly strong and



No. 3 Section of nut No. 2 made from bundled scrap.



No. 4 Section of nut No. 3 made from bundled scrap.

FIG. 3 (Continued)

tough. Photomicrographs Nos. 2, 3, and 4 of the same figure represent the structures of wrought-iron nuts Nos. 1, 2, and 3. As may be easily seen, the metal is made up of alternate layers of steel and wrought iron, the mottled-appearing portion being the steely part and the other the wrought iron. Such a condition is very undesirable on account of non-uniformity, and satisfactory physical properties could hardly be expected. Material of this nature is made by bundling wrought iron and steel scrap together, heating the bundles up to a welding temperature, and rolling the plastic mass out into merchant bars. Such stock is inferior to a

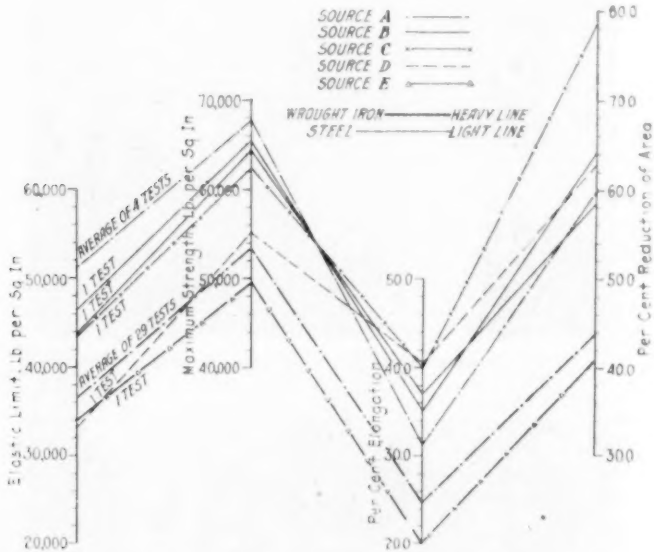


FIG. 4 COMPARISON OF TENSILE PROPERTIES OF MILD-CARBON-STEEL AND WROUGHT-IRON BOLTS

good grade of wrought iron. It is quite customary to use a material in the nuts which is somewhat inferior to that used in the bodies of the bolts, but the use of such poor stock as in this case is certainly to be condemned.

6 The argument which is usually advanced in favor of the use of wrought iron is that it exhibits greater ductility than any type of steel. In order to check up this question, along with some others which will be mentioned later, tensile tests were made upon a series of 71 steam-line and valve bolts. Each bolt was examined microscopically to determine the material from which it had been made, and it was found that nearly a third of them had been made from wrought iron. The results of these tests are shown in Figs. 4 and

5, where it may be seen that the average ductility of the wrought-iron bolts as shown by the percentage elongation and percentage reduction of area is about 18 per cent less than that of the mild-steel bolts, and no better than that of the high-carbon and heat-treated steel bolts which possess much greater elastic limits and tensile strengths. Table 2 contains the complete data from which Figs. 4 and 5 were prepared.

Steel vs. Screw Stock

7 Another situation which has been discovered in this investigation is the use of screw stock as a material from which to manu-

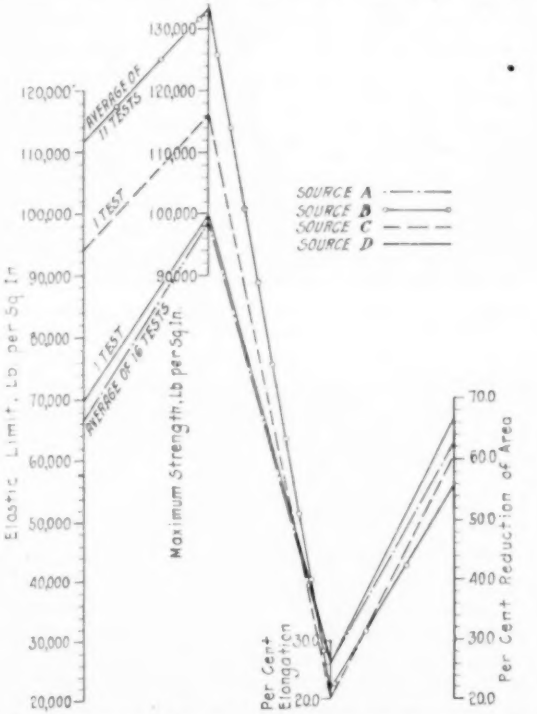


Fig. 5 Comparison of Tensile Properties of High-Carbon and Heat-Treated Steels

facture nuts. No instances were found where screw stock had been used in making the bodies of bolts, but a great many nuts were found to have been made from this material. The author cannot help but express his conviction that the use of this material is more dangerous, because of its treacherous brittleness, than the use of wrought iron.

8 It might be well to develop this proposition through a specific example. Two nuts were submitted which had cracked while being tightened. The chemical analyses of these two nuts were as follows:

	Carbon	Sulphur	Phosphorus
Nut No. 4.....	0.16	0.093	0.108
Nut No. 5.....	0.17	0.097	0.111

9 It may easily be seen from the analysis that these nuts were made from screw stock. The sulphur and phosphorus contents are just about double what they should be for freedom from brittleness. These analyses alone would condemn these nuts, but a consideration of the photomicrographs brings to light a further reason for their failure. Photomicrograph No. 1 of Fig. 6 represents an etched longitudinal section of nut No. 5. The dark areas are pearlite, which is the carbon-carrying constituent. The light-colored crystals are free ferrite. This structure is normal for screw stock. Photomicrograph No. 2 of the same figure is an unetched longitudinal section of nut No. 5. The small dark patches at the top are slag streaks, while the large dark portions at the bottom of the sections are pits or spongy spots. Such a condition represents a very poor metal even for screw stock, and it is very easy to see why these nuts cracked when a wrench was applied to them.

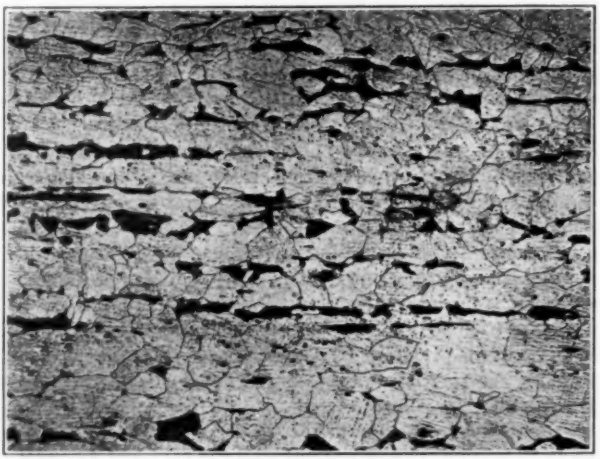
10 In a lot of bolts and nuts which was recently examined it was found that two-thirds of the nuts were made from a low-chrome steel while the other one-third were made from screw stock. Almost invariably a bolt which carried a screw-stock nut on one end carried a chrome-steel nut on the other end. Such practice is at best inconsistent. It might be termed "sending out a boy to do a man's work." Screw stock is not a satisfactory material for steam-line-bolt nuts.

The Best Carbon Content of Bolts

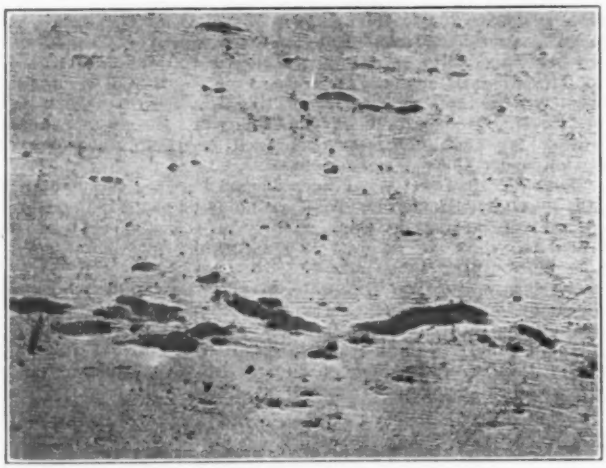
11 It is well known that one of the first requisites of good bolting is a sufficient amount of ductility. This is just as important as the possession of sufficient strength. As has been pointed out, it is probably on account of a supposed superiority in this respect that wrought iron is used so much as bolting. When, however, a bolting material fails from lack of strength, there is danger of going to the other extreme and increasing strength at the expense of ductility. Reference to Fig. 5 illustrates this point. The percentage elongation of the high-carbon and heat-treated bolts is about the same as that of wrought iron (Fig. 4), while the reduction of area averages slightly less that that of the mild-steel samples. While the high-strength bolts of Fig. 5 are entirely satisfactory, they represent the tendency mentioned, and were the analyses and heat treatment varied in such a way that the tensile strength would be increased notably, a point would be reached at last where

ductility would drop to a dangerously low point. Lack of ductility, of course, means brittleness.

12 It is a natural conclusion from what has just been said that the best carbon content for bolting steel lies somewhere between



No. 1 Etched section, showing dark crystals of pearlite in a mass of ferrite. Note banded structure, which is indicative of poor strength.



No. 2 Unetched section showing slag and pits.

FIG. 6 PHOTOMICROGRAPHS OF LONGITUDINAL SECTIONS OF NUT NO. 5
(Etch, 4 per cent picric-nitric acid mixture; magnification, × 100.)

0.10 per cent and 0.6 per cent. It would seem that the range from about 0.20 per cent to 0.45 per cent would yield the best combination of tensile strength and ductility in either the heat-treated or non-heat-treated condition.

Standard Procedure a Necessity

13 It would be possible to multiply the foregoing examples many times, but the result would be only to emphasize what is already evident, namely the lack of any uniform and consistent procedure in regard to the type of bolts which are used in the construction of steam-power-plant equipment. It is felt that it is unnecessary to argue in favor of the adoption of some standard procedure, for the thing seems obvious. One power plant, recognizing the necessity for covering this material with a rigid specification, has drawn up and adopted a specification under which it has been buying and inspecting bolts. It will not be included in this paper, for it is proposed to present for consideration of the Society the tentative specifications for high-temperature-alloy bolting which have been prepared by Committee A-1-22 of the American Society for Testing Materials. These specifications are here included in full.

TENTATIVE SPECIFICATIONS FOR ALLOY-STEEL BOLTING MATERIAL FOR HIGH-TEMPERATURE SERVICE

1 *Material Covered.* (*a*) These specifications cover alloy-steel bolting material for valves, flanges, and fittings for temperatures from 400 to 750 deg. fahr. (205 to 400 deg. cent.), inclusive. The term "bolting material" as used in these specifications covers rolled, forged or cold-drawn bars, and bolts, screws, and studs.

(*b*) Three classes of material are covered, A, B and C, classified in accordance with their physical properties as specified in Section 10.

(*c*) Nickel, chrome-nickel, chrome-vanadium, chrome-manganese, or any other types of alloy steel approved by the purchaser may be submitted under these specifications.

2 *Basis of Purchase.* When agreed upon between the manufacturer and the purchaser, Brinell hardness tests may be made to determine the acceptance of bolting material in lieu of the tension tests herein specified, but shall not be used as a basis for rejection without confirming tension tests being made.

3 *Certification of Test for Bolts, Screws, and Studs.* Unless otherwise specified on the order, a certification that bolts, screws, and studs for valves and other fittings intended for stock and other purposes requiring assembly in the manufacturer's plant conform to the requirements of these specifications, shall be accepted in lieu of the tests herein specified.

I — Manufacture

4 *Process.* The steel shall be made by one or more of the following processes: open-hearth, electric-furnace, or crucible.

5 *Discard.* A sufficient discard shall be made from each ingot to secure freedom from injurious piping and undue segregation.

6 *Heat Treatment.* (*a*) Heat treatment shall consist of quenching and tempering.

(*b*) Immediately after rolling or forging, the bolting material shall be allowed to cool to a temperature below the critical range under suitable conditions to prevent injury by too rapid cooling. For quenching and tempering, the material shall then be uniformly reheated to the proper temperature to refine the grain (a group thus reheated being known as a "quenching charge") and quenched in some medium under substantially uniform conditions for each quenching charge. The material shall then be uniformly reheated to the proper temperature for tempering or "drawing back" (a group thus reheated being known as a "tempering charge") and allowed to cool uniformly.

II — Chemical Properties and Tests

7 *Chemical Composition.* (*a*) The steel shall conform to the following requirements as to chemical composition:

Phosphorus, per cent	not over 0.04
Sulphur, per cent	not over 0.05

(*b*) The composition of the steel furnished under these specifications, other than phosphorus and sulphur, shall be agreed upon by the manufacturer and the purchaser.

Note: It is recommended that the carbon content shall not be less than 0.20 nor more than 0.45 per cent, and that the carbon ranges shall be 0.10 per cent of carbon.

8 *Ladle Analyses.* An analysis of each melt of steel shall be made by the steel manufacturer to determine the percentages of the elements present. This analysis shall be made from a test ingot taken during the pouring of the melt. The chemical composition thus determined shall be reported to the purchaser or his representative, and shall conform to the requirements specified in Section 7.

9 *Check Analyses.* Analyses may be made by the purchaser from samples representing the bolting material. The chemical composition thus determined shall conform to the requirements specified in Section 7, except that the rejection limit for phosphorus shall be 0.05 per cent and for sulphur 0.055 per cent.

III — Physical Properties and Tests

10 *Tension Tests.* (*a*) The bolting material, after final heat treatment, shall conform to the following requirements as to tensile properties:

	Class A	Class B	Class C
Tensile strength, lb. per sq. in.	95,000-115,000	105,000-125,000	not less than 125,000
Yield point, min., lb. per sq. in.	70,000	80,000	105,000
Elongation in 2 in., min., per cent	20	20	16
Reduction of area, min., per cent	50	50	50

(*b*) The yield point shall be determined by the drop of the beam of the testing machine or by the use of dividers.

11 *Brinell Hardness Tests.* (*a*) When Brinell hardness tests are made, the bolting material, after final heat treatment, shall conform to the following requirements:

	Brinell hardness number
Class A	190–250
Class B	210–270
Class C	260–320

(*b*) The Brinell hardness test shall be made in accordance with the Tentative Methods of Brinell Hardness Testing of Metallic Materials

(Serial Designation: E 10-25T) of the American Society for Testing Materials.[1]

12 *Tension Test Specimens.* (*a*) The tension test specimens taken from the bolting material shall conform to the dimensions shown in Fig. 1, except as specified in Paragraph (*b*). The ends shall be of a form to fit the holders of the testing machine in such a way that the load shall be axial.

(*b*) In the case of small sections which will not permit of taking the standard tension-test specimen specified in Paragraph (*a*), the tension-test specimen shall be as large as feasible and its dimensions shall be proportional to those shown in Fig. 7 (Fig. 1 of the specifications). The gage length for measuring elongation shall be four times the diameter of the specimen.[2]

13 *Number of Tests.* (*a*) Except as specified in Paragraph (*b*), one tension test shall be made for each lot of 100 pieces or less of bolting material from each tempering charge. If more than one quenching charge is represented in a tempering charge, one tension test shall be made from each lot of 100 pieces or less from each quenching charge. If more than one melt is represented in a quenching charge, one tension test shall be made from each lot of 100 pieces or less from each melt.

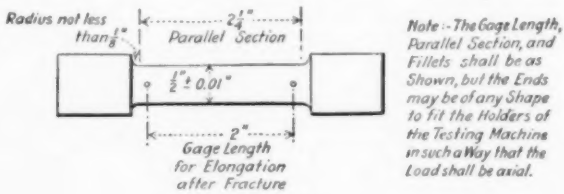

Radius not less than 1/8"
2 1/4"
Parallel Section
1/2" ± 0.01"
2"
Gage Length for Elongation after Fracture
Note:-The Gage Length, Parallel Section, and Fillets shall be as Shown, but the Ends may be of any Shape to fit the Holders of the Testing Machine in such a Way that the Load shall be axial.

FIG. 7

(*b*) Unless required on the order, tension tests shall not be made on an order of less than 100 bolts, screws, or studs; in which case acceptance shall be based upon the provisions in Sections 2 or 3.

(*c*) If any test specimen shows defective machining or develops flaws, it may be discarded and another specimen substituted.

(*d*) If the percentage of elongation of any test specimen is less than that specified in Section 10 (*a*) and any part of the fracture is outside of the middle three-fourths of the gage length, as indicated by scribe scratches marked on the specimen before testing, a retest shall be allowed.

14 *Retests.* If the results of the physical tests of any lot of bolting material do not conform to the requirements specified, the manufacturer may retreat such lot one or more times and retests shall be made as specified in Section 13 (*a*).

IV — WORKMANSHIP AND FINISH

15 *Finish.* The bolting material shall be free from injurious defects and shall have a workmanlike finish.

16 *Nuts and Washers.* Bolts and studs shall be equipped with cold-punched semi-finished carbon-steel nuts of U. S. Standard rough

[1] *Proceedings*, Am. Soc. Testing Matls., vol. 25, Part I (1925).

[2] These requirements are in accordance with the provisions of Section 5 of the Tentative Methods of Tension Testing of Metallic Materials (Serial Designation: E 8-25T). *Proceedings*, Am. Soc. Testing Matls., vol. 25, Part I (1925).

dimensions, chamfered and trimmed. If washers are used under nuts they shall be of forged or rolled steel. All nuts and washers shall be free from injurious defects and shall have a workmanlike finish, and shall conform to the chemical requirements for phosphorus and sulphur specified in Sections 7 (*a*) and 9.

17 *Threads.* U. S. Standard threads shall be used, except that no stud shall have less than eight threads per inch.

V —Marking

18 *Identification Marks.* Identification marks to be stamped on bolting material shall be agreed upon between the manufacturer and the purchaser.

VI — Inspection and Rejection

19 *Inspection.* (*a*) The inspector representing the purchaser shall have free entry, at all times while work on the contract of the purchaser is being performed, to all parts of the manufacturer's works which concern the manufacture of the bolting material ordered. The manufacturer shall afford the inspector, without charge, all reasonable facilities to satisfy him that the bolting material is being furnished in accordance with these specifications. Tests and inspection at the place of manufacture shall be made prior to shipment.

(*b*) The purchaser may make the tests to govern the acceptance or rejection of the bolting material in his own laboratory or elsewhere. Such tests, however, shall be made at the expense of the purchaser.

(*c*) Tests and inspection shall be so conducted as not to interfere unnecessarily with the operation of the works.

20 *Rejection.* (*a*) Unless otherwise specified, any rejection based on tests made in accordance with Section 19 (*b*) shall be reported within five working days from the receipt of samples.

(*b*) Bolting material which shows injurious defects while being finished by the purchaser will be rejected and the manufacturer shall be notified.

21 *Rehearing.* Samples tested in accordance with Section 19 (*b*), which represent rejected bolting material, shall be preserved for two weeks from the date of the test report. In case of dissatisfaction with the results of the test, the manufacurer may make claim for a rehearing within that time.

14 The rigid adoption of this specification will eliminate all the objectionable conditions and materials which have been referred to earlier in this paper. Sections 4, 7, and 16 eliminate the possibility of using wrought iron or screw stock as material for either bolts or nuts, and also limit the carbon content of the steel used to a range where ductility will not be too low on finished bolts.

15 It is mentioned as a final thought that, while it may be perfectly satisfactory to use high-quality carbon-steel bolting on the same job with alloy-steel bolting, there is always the danger of mixing up the two types; and since the difference in price is quite small, it is recommended that alloy bolting be used on all power-plant work.

DISCUSSION

L. E. THOMAS.[1] There is much misunderstanding regarding wrought iron, and portions of the paper indicate that there is some misunderstanding on the part of the author. Genuine wrought iron is made entirely from puddled pig iron. It is entirely free from any admixtures of iron or steel scrap, no scrap whatever being used in the manufacture, except the mill's own crop ends. Much

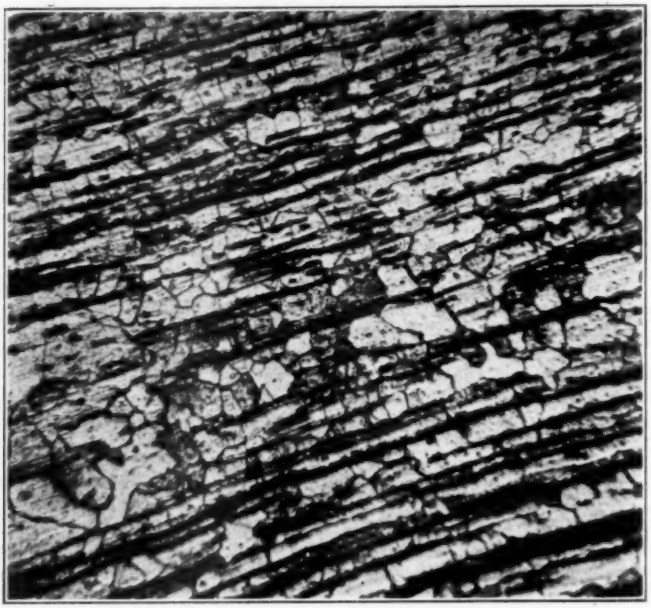


FIG. 8 PHOTOMICROGRAPH OF GENUINE WROUGHT IRON
(Magnification, × 100)

material is made and sold as wrought iron that is made entirely or in part from iron or steel scrap. Hence, it is necessary to refer to wrought iron made entirely of puddled pig iron as genuine wrought iron.

Upon examination genuine wrought iron is found to be a composite mixture of ferrite and non-conducting, non-corroding silicious slag resembling glass in its properties. In rolling this material, the slag and the iron are drawn out in long fibers, as shown in Fig. 8, which are distributed throughout the structure. Genuine wrought iron thus has a fibrous structure like a piece of hickory,

[1] President, Reading Iron Co., Reading, Pa.

which gives it superior fatigue-resisting qualities. This characteristic makes it particularly suitable for service where the material is subject to severe and sudden shocks, vibratory strains, and repeated reversals of stresses. For this reason it is almost universally used by the railroads for such parts as staybolts, engine bolts, draw bars, equalizers, air and water pipe, and other parts subject to shocks and vibratory strains.

Not many years ago all bolts and nuts and many other products were made from genuine wrought iron. The advent of steel, a cheaper material to manufacture, and for many purposes fully as satisfactory as wrought iron, led to the use of steel for the manufacture of bolts and nuts. In order to meet this competition, bolt and nut manufacturers have used an inferior quality of wrought iron, made from either busheled or piled scrap. The bolts and nuts described by the author as wrought iron were made of this inferior material. For many years after the introduction of bolts made of this cheaper material many consumers, even the railroads, refused to use them, especially for track work. This was due to the fact that variations in temperature set up stresses in the track that caused the steel bolts to snap off in the threads. Steel is crystalline in structure as shown in photomicrograph No. 1, Fig. 3, and it will be noted in photomicrographs Nos. 2, 3, and 4, which are of bolts made from bundled scrap, that they likewise show a crystalline structure, although, of course, not as fine or uniform as photomicrograph No. 1, which is from a low-carbon steel bolt. Bolts made from steel and bundled steel scrap are crystalline in structure, and fractures in this material are due to slip lines being set up at the point of maximum stress in the cleavage planes of the crystals, and also to the unequal distribution of the stresses among the crystals. Bolts made of genuine wrought iron are not affected by varying conditions, as the material is fibrous in structure and resistant to fatigue and repeated stresses.

The superior fatigue-resisting qualities of wrought iron as compared with steel are shown in Technological Paper No. 289, published by the Bureau of Standards. The genuine wrought iron used in the tests showed a carbon content of 0.03 per cent and manganese 0.02 per cent, while the steel showed 0.25 per cent carbon. This is the carbon content in the steel recommended by the author in Par. 12. The physical properties of the two materials are as follows:

	Yield point, lb. per sq. in.	Tensile strength, lb. per sq. in.	Elongation in 2 in.	Reduction of area
Wrought iron	29,250	47,000	33.7 per cent	46.4 per cent
Steel	35,800	62,600	37.5 per cent	57.2 per cent

In making the sudden impact test an Izod pendulum-type machine with a capacity of 120 ft-lb. was used, the striking velocity of the hammer being 12.44 ft. per sec. With the standard V-notch

cut in the plane of rolling, the total energy absorbed by the wrought iron was 58.40 ft-lb., while the steel absorbed only 16.33 ft-lb.

With the notch at right angles to the plane of rolling, similar to the threads on bolts, the energy absorbed by wrought iron was 39.37 ft-lb., and by steel, 15.22 ft-lb. The figures represent an average of a number of tests on both iron and steel. In each test the steel broke in two, while all the wrought iron test-pieces merely bent over and disclosed a good fibrous structure. Tests of this character show conclusively that wrought iron by reason of its superior qualities should always be used in installations subjected to sudden shocks or high stresses produced by any suddenly applied force.

The writer does not agree with the author that the elastic limit and tensile strength should be at least 50 per cent more than they are. The American Society for Testing Materials at its annual meeting in June, 1925, decided to lower the tensile strength of staybolt iron, the minimum now being 47,000 lb. per sq. in. The majority of the railroads prefer staybolt iron of an even lower

TABLE 3 TESTS OF STAYBOLT IRON

Test piece			Elastic limit		Ultimate strength			
No.	Diam., in.	Area, sq. in.	Load applied, lb.	Lb. unit stress per sq. in.	Load applied, lb.	Lb. unit stress per sq. in.	Per cent elongation in 2 in.	Per cent reduction of area
1¼	Rd. 1.238	1.243	35,500	29,360	58,200	46,620	31	53.3
"	1.258	1.243	36,500	29,360	58,000	46,660	33.5	52.2
"	1.256	1.229	35,200	28,640	57,200	46,540	32.5	52.2
"	1.258	1.243	35,700	28,720	57,200	46,020	32	51.6
"	1.256	1.229	36,100	29,370	57,700	46,950	31	50.5
"	1.258	1.243	36,000	28,960	58,500	47,060	31	51.5

strength and are accepting material of a tensile strength of 46,000 lb. per sq. in. It has been found that genuine wrought iron of these lower physical properties possesses sufficient strength to withstand the most severe service to which it may be subjected. Table 3 presents tests of genuine wrought iron that were accepted by one railroad as entirely satisfactory material for use in locomotives.

The author states in Par. 6 " the argument which is usually advanced in favor of the use of wrought iron is that it exhibits greater ductility than any type of steel," and disputes this claim by presenting data in Table 2 that show no superior qualities of the wrought-iron bolts that were tested. The photomicrograph of the so-called wrought-iron bolts shows that they were made from scrap and were not genuine wrought iron. On the other hand, it is true that soft steel of the same tensile strength as wrought iron will show greater elongation and reduction of area than genuine wrought iron. This is true for laboratory tests taken in specimens 8 or 10 in. long; but it is not true in service when taken in full-

length eye-bars. Engineers recognize that it possesses the necessary qualities imparted to it by its fibrous structure to warrant its use in all the important parts of locomotives and cars subject to severe and sudden shocks. In other words, while the initial physical properties of genuine wrought iron are not superior to those of soft steel, it will not deteriorate in service. Therefore, genuine wrought-iron bolts are superior to steel bolts, excepting perhaps those of special heat-treated steel. If the bolts are to be located in a damp, corrosive atmosphere, there is no doubt that the wrought iron will prove more resistant to corrosion. This superior corrosion-resisting property of genuine wrought iron is imparted to it by fine, interlocking non-corroding slag bands, which everywhere throughout the structure obstruct the progress of corrosion attack.

L. W. Spring.[1] The subject of bolting for high pressures and temperatures seems yet to be a little more of an open one than it should be, as is evidenced by the results shown in this interesting paper. It usually is considered that bolting is one of the weakest parts of an installation. This is because small bolt circles are desirable, and small bolt circles preclude excess of bolts, both as to number and size. Therefore, ordinarily the best bolts possible should be provided.

The objections of the author of this paper to wrought-iron bolts are valid. The material is never strong. Some designing engineers argue for steel bolts with only 60,000 to 80,000 lb. per sq. in. yield point, desiring, mainly, high ductility. But how useful are bolts if they stretch in service, and is their high ductility of service in keeping a plant going? Many others think that where stresses warrant it high-yield-point bolts, i.e., much stronger and stiffer ones which never get stressed to the yield point, are desirable. Such are the new Class C alloy-steel bolts of A.S.T.M. Specification A 96-26T with a yield point of 105,000 lb. per sq. in. minimum. Naturally such bolts have not as great ductility as those of lower strength. But, primarily, ductility is of service only after the yield point has been exceeded, and if we do not exceed the yield point, is not material of 16 per cent or even 14 per cent elongation as safe as that of 20 per cent or even 25 per cent? Moreover, elongation increases with temperature, quite rapidly above 450 or 500 deg. fahr., so that at 700 deg. fahr., ordinarily, it is from 25 to 50 per cent greater than at normal temperature. Endurance values of steel increase as tensile and Brinell values increase, so under weaving action or other reversing stresses, high-strength bolting should be of advantage. Impact values, however, usually decrease as tensile strengths increase, but are higher at temperatures up to 700 deg. fahr., or more, than at normal temperature.

[1] Chief Chemist and Metallurgist, Crane Company, Chicago, Ill.

It should be said, further, that headed bolts often suffer non-uniformity over their length because of dissimilar heat conditions and work in the heading operation.

It would seem that the safest article is the stud bolt cut from high-grade rolled steel bar and threaded on both ends either before

TABLE 4 ANALYSIS AND PHYSICAL PROPERTIES OF CLASS C ALLOY STUD STEEL (HEAT-TREATED)*

Size, in.	Tensile strength, lb. per sq. in.	Yield point, lb. per sq. in.	Per cent elongation in 2 in.	Per cent reduction of area	Izod test	Brinell hardness number
1	120,700	98,200	20.0	66.2	78	253
1	128,300	109,500	21.0	63.7	75	269
1	125,300	117,000	20.0	62.5	69	282
$\frac{1}{4}$	133,000	120,000	19.0	63.4	66	288
$1\frac{1}{8}$	139,000	126,000	18.0	62.5	64	293
1	141,000	118,500	17.5	55.0	45	302
1	140,300	113,000	18.5	67.5	67	302
1	....	....	....	....	58	311
$\frac{3}{4}$	141,600	125,500	16.0	62.4	71	...
1	149,800	130,800	17.0	55.2	54	...
1	147,000	119,500	18.0	49.3	39	321
1	146,350	131,000	17.0	54.4	40	329
1	151,700	137,800	15.5	51.5	38	334
1	....	....	....	....	37	351

* Chrome-nickel and chrome-molybdenum steels of approximately 0.35 per cent carbon.

or after careful heat treatment. Stud bolts of this Class C material (Table 4) have been in use in many installations under severe service conditions for a number of years with very satisfactory results, whereas in similar service, materials with only 60,000 to 80,000 lb. per. sq. in. yield point have failed, due to stretching.

W. OBERHUBER [1] and J. B. ABELE.[2] The material for bolts should most assuredly receive the same careful attention that is paid to other material entering into power-plant construction. The author advocates the use of heat-treated alloy-steel bolts for high-pressure steam lines and rightly so, more particularly because of the high elastic limit that this material possesses.

With the use of ordinary carbon-steel bolts with an elastic limit somewhere around 30,000 lb. per sq. in., it is common to find that smaller sizes of bolts are stressed to this point or beyond. While bolts should never be stressed up to their elastic limit, it is particularly important that this point should never be reached in the case of bolts in a steam line subjected to wide variations in temperature if the joints are to remain tight. Any attempt to maintain tight joints with bolts stressed beyond their elastic limit will result in their ultimate failure. When pulling up bolts in a steam line they should never be stressed to anywhere near their elastic limit, for the reason that the application of heat with the attendant increase in temperature of the pipe will put an additional stress in the bolt,

[1] Supt. Maintenance Dept., Philadelphia Electric Co., Philadelphia, Pa. Assoc. Mem. A.S.M.E.

[2] Research Dept., Philadelphia Electric Co., Philadelphia, Pa.

before the latter has reached the temperature of the pipe. For this reason alone the property of high elastic limit in a heat-treated alloy-steel bolt is a distinct advantage.

There is no practical means outside of the laboratory for determining the stress set up in a bolt. Any attempt to limit the length of wrenches to be used is hardly practical, for too often a pipe-fitter finds that a piece of pipe applied to the end of a wrench makes lighter work for him. Some very interesting tests were made, however, in our laboratory to determine the amount of pull on the end of a wrench required to stress the bolts up to certain points. A standard 100,000-lb. Riehle testing machine was used, and the weight on the beam was set at the point representing the load on

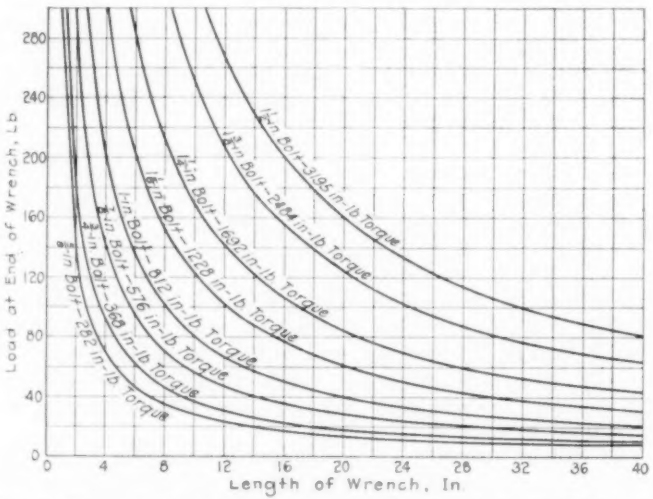

Load at End of Wrench, Lb
Length of Wrench, In.
1½in Bolt-3195 in-lb Torque
1¼in Bolt-2484 in-lb Torque
1in Bolt-1692 in-lb Torque
⅞in Bolt-1228 in-lb Torque
¾in Bolt-812 in-lb Torque
⅝in Bolt-576 in-lb Torque
½in Bolt-360 in-lb Torque
⅜in Bolt-282 in-lb Torque

FIG. 9 LINES OF CONSTANT TORQUE FOR VARIOUS SIZES OF BOLTS AND LENGTHS OF WRENCHES

(Length of wrench and load at end of wrench to produce a fiber stress of 10,000 lb. per sq. in. at root of thread on various-sized bolts. Fiber stress in bolt increases directly as the load at end of wrench or directly as the length of wrench.)

the bolt required to stress it to a certain figure. The nut was pulled up with a wrench, to the end of which a spring balance was applied. The load recorded by the balance was noted when the beam tipped. All sizes of bolts from $\frac{5}{8}$ in. to $1\frac{1}{2}$ in. were tested. It was found that an average of about 88 per cent of the work done was dissipated in friction. The remaining 12 per cent was used in stressing the bolt. It was also found that the fiber stress in a bolt increases directly as the load at the end of a wrench, and directly as the length of the wrench. From these tests the chart, Fig. 9, has been prepared to show the lines of constant torque for the various sizes of bolts when stressed to 10,000 lb.

per sq. in. The load required may be found if the length of wrench is known or vice versa. As there is no practical means of telling just what load is put on the end of a wrench, it is more or less a matter of conjecture as to the amount of stress set up in a bolt. A heat-treated alloy-steel bolt with a high elastic limit, however, gives a greater margin of safety than an ordinary carbon-steel bolt.

We recommend rigid inspection of all bolts in vital places and their periodic removal, to heat-treat, test and examine for any change in structure and physical strength. Some carbon-steel bolts that were in service for about 10 years were removed and inspected. They showed a marked crystalline growth somewhat similar to what is known as Stead's brittleness as referred to by Prof. Bradley Stoughton in his Metallurgy of Iron and Steel.

We agree with the author that the use of screw stock for nuts is to be condemned on account of its brittleness. We were forced to return a lot of nuts made of screw stock because of their failure, and in their place we substituted cold-punched carbon-steel nuts which we believe will give satisfactory service.

We find that a quicker method of testing a nut is by crushing it under a steam hammer. If the nut shows upon inspection that it has sufficient ductility and does not fracture, we conclude that the lot it represents is satisfactory for service.

We believe that the large quantities of slag inclusions in wrought iron, as well as its low carbon content, has a greater tendency for grain growth and makes it objectionable for use as bolting material. Wrought iron also has a low elastic limit which is another objectionable feature. We consider steel with a carbon content of 0.60 per cent as too high and that a carbon content of about 0.30 per cent is better. We also agree with the author that ductility is just as essential as strength.

The writers' company has a standard specification for bolting material and it agrees substantially with the specifications referred to in the paper, except that we specify chrome-vanadium steel, although any heat-treated alloy steel that meets the physical requirements and the chemical requirements as to phosphorus and sulphur content might prove quite as satisfactory and reliable. We specify vanadium for the reason that it increases the tensile strength and raises the elastic ratio without materially lowering the ductility. Vanadium steels are not subjected to excessive grain growths until a temperature of 1700 deg. fahr. is reached. This property also enables vanadium to maintain a larger proportion of strength and hardness at elevated temperatures.

We use studs exclusively on all high-pressure steam lines, and all alloy-steel studs are stamped on the end to distinguish them from all ordinary studs. One nut is screwed on and peened over to insure a full length of stud in the nut. We question Par. 17 of the tentative specifications on the advisability of using studs with threads that are not standard as mentioned in those specifica-

tions. We also specify a tolerance for the diameter of the bars for studs, and the amount of out-of-round permissible. We believe that electric or crucible steel is preferable to open-hearth or bessemer steel on account of its uniformity of structure.

Our heat treatment of the stud material consisted of heating to 1600 deg. fahr., quenching in oil, and drawing to 1225 deg. fahr. and then allowing it to cool slowly.

Our conclusions are:

a While we agree with the author, we believe that vanadium steel, manufactured by the electric or crucible process, is preferable, on account of the more positive assurance of better steel.

b We believe that much work is yet to be done to discover the causes of the brittleness in bolt material after it has been subjected to temperatures and strains similar to steam-pipe conditions.

c We are also of the opinion that a system of length in wrenches should be worked out, with a normal pull of 40 lb. on the end of the wrench by one man, instead of the usual practice of two men pulling on one wrench, or of using a piece of pipe for extending the length of wrenches.

N. L. MOCHEL.[1] Much has been said regarding the use of wrought iron for bolting and nuts. Referring to the results of two tests given in Table 1, it is stated that "the elastic limit and ultimate strength should be at least 50 per cent higher than they are." It is assumed that reference is made to the application requiring a material of at least 50 per cent higher properties, and not that the wrought iron should have given those higher results. While referring to "elastic limit," we would suggest that the results in Tables 1 and 2 for "elastic limit" seem high, and it appears that "yield point" is really intended. If so, we would offer the opinion that "yield point" should be referred to as "yield point," as wrong impressions may be given by referring to the same as "elastic limit."

A very large amount of wrought iron has been used for bolts and nuts, and successfully used in most cases. Present-day knowledge and experiences, however, point to steels being the more suitable for the purpose in general. As to slag inclusions, all wrought iron contains them, and experiences have shown that shipments of the best grades of imported iron are very apt to contain numerous bars having the enormous slag inclusions referred to by the author. He refers to that argument usually advanced in favor of wrought iron, that it exhibits greater ductility than any type of steel. We confirm his findings that such is not true, and that a good mild steel, such as the Navy Class B bolt steel, has superior properties in this respect, as well as higher yield point and tensile strength.

[1] Metallurgical Engineer, Westinghouse Elec. & Mfg. Co., South Philadelphia Works, Philadelphia, Pa.

The author's tests were made on machined test specimens of the approved type. Had tests been made on actual studs or bolts in their natural condition, with nuts in place, and load applied through the nuts, failure would have occurred in the threads for both steel and iron, but even greater differences in elongation and reduction of area would have been observed.

In a recent brief study we found a very large proportion of commercial nuts to be made of wrought iron, notwithstanding the general impression that steel had almost entirely replaced iron for the purpose. Nuts of wrought iron have been known to be so greatly weakened by large slag areas as to split while being tightened. Evidence also exists that wrought-iron nuts are more apt to seize or gall on cast-steel faces under elevated temperature conditions than are steel nuts.

The statement that while screw stock had been used for nuts, no screw stock had been used for the bolts proper, is surprising. Much evidence exists that a large proportion of commercial bolts and studs on the market today are made from bessemer or open-hearth screw stock, high in phosphorus and sulphur, or in sulphur alone. In general, they are brittle and unsatisfactory for the purpose. It is dangerous practice to purchase bolts and studs indiscriminately in the market today.

Par. 12 suggests that a carbon range of from about 0.20 per cent to 0.45 per cent would yield the best combination of strength and ductility in either the heat-treated or rolled condition. We would elaborate slightly on this and say that any 0.10 per cent range of carbon from 0.20 per cent to 0.45 per cent should give good results. We suggest this in the interests of uniformity in any given lot of bolting material, as uniformity of that degree is necessary from nearly any standpoint of heat treatment or application.

Par. 18 of the specifications touches on the important matter of marking. Bolts made of special materials or for special applications should always be so marked or be of such different design as to prevent their being mixed up and used in the wrong place. There are many instances where good grades of carbon steel are entirely satisfactory as bolting material; and it has been observed in the tentative standards prepared by the Sectional Committee on the Standardization of Pipe Flanges and Fittings organized under the procedure of the American Engineering Standards Committee that the use of good carbon steel has been recognized.

W. W. Boyd.[1] As the steel bolts come from the header, they are in various stages of hardness and softness due to the degree of heat at which they were worked. Most bolts are air cooled, although some are quenched in water as rapidly as forged. However, for

[1] Buffalo Bolt Co., North Tonawanda, N. Y. Assoc-Mem. A.S.M.E.

a uniform product, all must be normalized, or annealed, under the same condition. Annealing consists in

a Heating the steel to some predetermined temperature

b Keeping the temperature constant at this predetermined point for a given length of time

c Cooling the steel according to some predetermined course to atmospheric temperature.

The purpose of annealing may involve any one of the following:

a To soften the steel so that it may meet certain physical requirements or be more easily machined

b To reduce internal stresses caused by rolling, forging, drawing, etc.

c To remove coarseness of grain and thus secure a more desirable combination of strength, elasticity, and ductility for resisting stresses to which it is to be subject in service.

Size of grain is regulated by heating to the proper temperature and regulating the rate and manner of cooling. Furnace cooling gives the maximum softness, ductility, elongation, and reduction, likewise the minimum tensile strength and elastic limit.

Hardening consists of two operations, viz., heating to a suitable temperature and suddenly, or rapidly, cooling. Whatever the structure of the grain before hardening, it is entirely changed and becomes totally different from that of unhardened steel.

Tempering or drawing back consists of reheating the metal after hardening to some temperature below the critical range, and may have for its primary object

a The regulation of the hardness and brittleness of the steel

b The toughening of it

c The release of the hardening strains.

The release of the hardening strains is usually accomplished by heating, in the tempering process, because heating even to the temperature of boiling water will relieve the strains to some extent.

Bolts in steam lines are subjected to temperatures well within the tempering ranges and are exposed to this temperature for long periods of time with but narrow variations. When steam lines are thoroughly insulated, the cooling amounts to nothing. When cooling does take place, the insulation prevents the rapid dissipation of heat, and we then have a condition similar to furnace annealing. Due to the continual drawing of temper, it is but a question of time when the physical effects of expertly heat-treated steel are nullified and the steel assumes a state similar to that of thoroughly annealed steel.

Heat treatment is necessary to develop the beneficial effects of chromium in steel, which is most active in responding to this treatment. Chromium is preëminently a hardening agent in steel. Unlike nickel, which merely dissolves in iron, chromium forms a carbide. Chrome-nickel steels of suitable composition appear to have combined in them the beneficial effects of both chrome and

nickel without the disadvantages which are inherent in the use of each one separately. In an annealed state, the physical properties of simple nickel or chrome steels, chrome-vanadium, or chrome-nickel steels are much superior to annealed plain carbon steels. Heat treatment, so far as refining the grain and relieving stresses due to forging is concerned, is of distinct benefit, but hardening and tempering are of somewhat doubtful value for steam-line bolts.

It is a curious fact that the records of tests of various grades of steels at high temperatures show a pronounced decrease in strength at about the 500 deg. fahr. point, with a natural increase in the elongation and reduction of area as the temperature increases.

In the manufacture of a certain class of machinery it was necessary to use hardened steel pieces inserted into castings which were afterward japanned and baked. These hardened carbon-steel pieces could not be touched by a file after they had been hardened, but after several hours in the japanning oven they were soft and useless. The solution of the problem was not to japan the inserts.

Coarse, granular structure due to working the metal below the proper forging temperature can be very much improved by normalizing, but where cracks have developed, heat treating is not a remedy.

V. T. MALCOLM.[1] Periodically troubles with bolts crop up in power-plant equipment, and cause much expense and worry due to the closing down of important units. The main problem as we see it, for the manufacturers of bolts, is to obtain steel that has uniform chemical and physical properties, heat after heat, and when this has become definitely established a certain standardized heat treatment may be determined. This should not be varied under any circumstances after it has been proved to be the best for the steel at hand.

Of course, all metallurgists do not agree upon certain specifications, but regardless of this fact, we have today certain specifications, as the tentative standards A-96-25 of the American Society for Testing Materials, that demand a high class of steel for use in power-plant work. It must be realized that there is no precedent that has been established for this class of work, as the demands of today are far more exacting and the service more severe, than at any time during the past.

One of the most common expressions heard in the accounting for failures of bolts, is that failure is due to crystallization of the steel. This idea is erroneous, and undoubtedly originated from the appearance of a fracture, and should be known as a "fatigue failure" — or, as Professor Homerberg calls it, a "progressive

[1] Metallurgical Engineer, The Chapman Valve Mfg. Co., Indian Orchard, Mass.

failure" — produced as the result of alternation of stresses, in which part of the fracture appears more or less smooth and the remainder granular or crystalline. If the bolts were subjected to a single stress they would fulfill these requirements indefinitely, provided the limited stress was not exceeded, but if subjected to alternating stresses, the bolt must ultimately fail. However, it is necessary in this connection to take into consideration not only the strain which results from external loading, but also the state of internal strains which have developed in the steel. This second factor is one that has been overlooked generally in engineering structures, and little consideration has been given to the influence of internal strains in the steel. These strains in our opinion are just as tangible as those which result from external loads and it is believed that they account for failures which have not been adequately explained by reason of the magnitude of external strains. In most cases these defects in bolt steels are due to strains set up in quenching and which are not entirely relieved by the tempering operation, or originally were so great that they cracked the steel. Strains may be also set up in non-heat-treated steel, due to the casting or rolling operation.

Steel in this state is an aggregate of crystals, and when the metal undergoes plastic deformation the crystals are deformed in the same general sense as the metal. This change of shape in the crystals is accompanied by a process of "slip" or "gliding" by which layers of crystals slide over one another, along certain gliding planes. When the slip is severe or is intensely localized, as when a single crystal is bent upon itself, the molecular disturbance at the slip surfaces increases, and permanent layers of amorphous metal are formed in each surface where slip occurs. The metal finally, as a result of extreme deformation, becomes a mass of crystal fragments imbedded in layers of relatively thick layers of amorphous metal which is accompanied by increased hardness of the steel, combined with great brittleness.

As this process continues, the crystals begin to lose their strength and ultimately the crack or flaw started from this condition works its way across the entire section until complete failure results. Quite frequently engineers, when ordering bolt steel, specify a soft carbon steel, without heat treatment, believing that the increased ductility obtained in soft steel will guard against fatigue. It has been demonstrated, however, that the ductility of steel in tension is not a criterion of the dynamic value of the steel. In the case of soft carbon-steel bolting, which is known to show good ductility in tension, we find that it has a low resistance when submitted to the impact test, and low resistance in impact certainly shows low cohesive properties of the metal itself, and therefore low fatigue value. It is therefore unfortunate that the impression still exists in some cases that good resistance to dynamic stresses may be assured in all cases by merely obtaining high ductility in

tension. This may be true of certain grades of steel but it is unsound for general application, especially for bolt steels.

One of the reasons advanced for the capricious behavior of steels high in phosphorus is that the same amount of phosphorus acts in different ways. In steel high in phosphorus the phosphorus is combined with iron as a ferro-phosphide, which in turn is alloyed with the residual iron. Consequently we have a direct alloy relation, generally more intensified locally, above the average condition, because of the marked tendency of the phosphorus to segregate in bands or streaks. Under repeated stress or shock, this results in cracks in these high-phosphorus zones, the crack subsequently travelling through the body of the steel, because of an unobstructed path through a crystalline mass.

Sulphur is present in steel in two forms, namely, iron sulphide and manganese sulphide, of which the latter is the more common. The sulphide of iron is present when sulphur is high and there is insufficient manganese present to react completely. The residual sulphide remains molten and is deposited along the grain boundaries of the metal. This forms an intercrystalline film which has very little strength, and when such metal is put under stress, rupture takes place with a conchoidal fracture along the grain boundaries. In like manner, when molten metal containing high sulphur is passing from the liquid to the solid state, strain is set up due to shrinkage, and the metal gives way along these films, causing shrinkage cracks. This iron sulphide spreads out in thin sheets when the metal has solidified and gives rise to dangerous zones of weakness when cold, as there is little cohesion between the crystalline grains, thus causing rupture at low loading when hot.

When sulphur is high in the form of manganese sulphide it often segregates, when combined with high phosphorus or slag, to form "ghosts" on the banded structure, which, of course, is very detrimental to the quality of the steel.

One of the greatest problems against which manufacturers of bolts have to guard is dirty steel, for the reason that dirty steel causes premature failure of bolts when in service. A series of endurance tests made recently on steel of this type has shown a great difference between sister bars of the same material handled in the same way and tested at the same nominal stress, except that one part was known to be dirty. It is also known that a surface notch or scratch greatly increases the local stress over the nominal calculated stress. Therefore, internal notches, such as those caused by inclusions or sonims, have the same effect. Users of steel carrying out the bend test have found surprisingly low ductility in steel containing inclusions, even though the tension bars may have been good. Therefore, it is important that clean steel be used in the manufacture of bolts, for once dirty steel has solidified, nothing can be done to remedy it. Almost every page of Giollitti's book emphasizes the effect of inclusions in steel, and

a large list of authorities can be cited to show that dirty steel is dangerous; but the situation is so obvious that proof is hardly needed.

Primary and secondary "pipes" are another source of danger in bolts and little need be said about them, except for the fact that the steel maker has failed to crop the original ingot sufficiently, and that the pipe left in the ingot has been rolled into the bar, causing the defect. This, of course, is a very dangerous source of defect.

It is, in our opinion, necessary to guard against the foregoing defects, and we have found it good practice to use the following safeguards:

a All steel entering the factory for the manufacture of bolts should be given a visual examination, and small sections cut from each bar and subjected to a hot etch to determine whether the steel contains any seams, cracks, or segregation.

b In addition each bar should be stamped with the heat number from the steel mill, showing the melt, and the chemical analysis should be checked, after which several bars should be cut and heat treated in accordance with standard practice; these bars should then be tested in tension and impact, and the fractures carefully examined.

c Any bolts at any time during the manufacture that are of doubtful quality should be given a Brinell hardness test in order to approximate the tensile strength, and if they do not meet certain requirements, the whole lot should be rejected.

We believe that with these precautions the consumer or user has little to worry about in the applications of bolts to his equipment.

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The Author. With reference to the points brought out in the discussion, there are a few in connection with which the author wishes to make his position clear and others whose importance he wishes to emphasize.

Mr. Thomas advanced the opinion that there was considerable misunderstanding concerning the types and properties of wrought iron and bundled scrap, and that the author had misunderstood some facts in that connection. The author has examined many samples of wrought iron and bundled scrap and has felt that there was little difficulty in identifying each type of material. The point to be stressed in this connection, is that there is considerable variation in the quality of puddled iron. Some irons are much

better than others, the chief difference being the amount and distribution of the slag. Bundled scrap always exhibits a sharply defined banded structure which is not found in puddled iron. Photomicrograph No. 3, Fig. 3, illustrates this. If steel scrap is used, there is no doubt of the identification and it is felt that a large part of the bolts which were examined in this investigation were made from puddled iron.

In the third paragraph of Mr. Thomas' discussion the statement is made that steel and bundled scrap are crystalline in nature while genuine wrought iron is fibrous in structure. If a study of Fig. 8 (a section of genuine wrought iron) is made, it can clearly be seen that the ferrite matrix of the sample is just as crystalline as any ferrite, and the author is unable to see why a fatigue fracture could not take place in this ferrite as readily as in any other ferrite. It would also appear that the junction of a slag ribbon and the surrounding ferrite might offer a good path for fracture in the direction of the slag ribbon.

Mr. Thomas refers to impact tests made by the Bureau of Standards (Technical Paper No. 289) as showing the superiority of wrought iron in shock resistance. It is to be noted that all the steels reported in that work were tested in the condition "as received from the mill." That means that the steel was not heat treated and may possibly have been in a coarse-grained condition. As a matter of fact the microsection of a nickel steel which was reported on looked to be decidedly coarse-grained. The great advantage in the use of steel lies in its ability to be treated in a variety of ways, thus making possible a large variation in properties. Any one of the steels treated in Technical Paper No. 289 could have had its impact value very materially improved by suitable mechanical or thermal means. In this same investigation, tests were made upon Armco iron, the analysis of which, neglecting slag, is almost identical with that of the wrought iron. The Armco iron is practically pure crystalline ferrite and it will be seen that the impact values of this material are consistently higher than those of the wrought iron which contains about 97 per cent ferrite. The conclusion, therefore, is unavoidable that the impact values of the wrought iron are due largely to the crystalline ferrite, rather than to its fibrous structure caused by the slag.

It is felt that Mr. Thomas has not kept clearly in mind the fact that bolts for power-plant construction only were under consideration. We should not compare procedure in that field with railroad work. Let the railroads reduce their tensile-strength requirement in staybolts if they wish. Power plants would be very foolish to accept low tensile strength in steam-line bolts. Because of the frequency of broken side rods on locomotives the author believes that it would be well for the railroads to give more attention to the question of adequate physical properties.

In the last paragraph of Mr. Thomas' discussion, the statement is made that laboratory tests show a greater elongation in the case of soft steel than in wrought iron, but that in full-length eyebars this does not hold true. If this statement is correct it would seem that our methods of physical testing are at fault. The author is unable to conceive how the presence of glassy slag can increase the ductility of ferrite.

Lastly, while he does not wish to enter an argument regarding the corrosion resistance of wrought iron, the author has yet to be shown how the ribbons of slag in wrought iron arrest the action of the corroding medium.

Mr. Spring very correctly stresses the importance of high strength as well as ductility. When one pauses to contemplate the enormous stresses which must exist in a large high-pressure steam line, it would seem that there would be little argument against the use of the strongest bolting material possible. The author also believes that headed bolts are inferior to stud bolts. At least he has noticed that their structure is not so good. Fig. 2 represents sections from two headed bolts.

Messrs. Oberhuber and Abele mention the tension in bolts at the time of erection as being of considerable importance. This is a point which is well worth considering and would merit a large amount of study. It doubtless is an important factor in the performance and life of the bolt.

Too much cannot be said regarding the possible development of brittleness in steam-line bolts. It is well known that in steel carrying from 0.05 to 0.15 per cent carbon, serious brittleness will result if cold stressing of the bolt is followed by heating to temperatures in the neighborhood of 750 deg. fahr. It is perhaps not so well realized that there is a critical stress which will cause the maximum grain growth and brittleness. Specification A-96-26T places the lower limit of carbon content at 0.20 per cent. Nothing lower than this should be used, particularly in the case of carbon steels.

Mr. Mochel raises the question as to whether "yield point" rather than elastic limit was meant in Tables 1 and 2. These points were determined by the "drop of the beam," and should probably be designated as Mr. Mochel suggests. Mr. Mochel states that even the best imported irons show slag inclusions. It has been the author's experience that imported puddled irons are much worse in this respect than those made in our own country. Mr. Mochel expressed surprise that screw stock had been used for nuts and not for bolts. In this connection it would be well again to point out that only bolts used in power-plant construction were examined. As Mr. Mochel intimates, there are hundreds of thousands of ordinary bolts made from screw stock, but from observations made in this investigation, it seems clear that the suppliers

had some doubts in regard to using screw stock for bolts in steam-line construction, but thought it might do for nuts.

Mr. Boyd emphasized the fact that bolts in steam lines are subjected to temperatures well within the tempering range and therefore might be softened to a certain extent. It is interesting in this connection to refer to a paper, by French and Tucker,[1] in which they state that "carbon and the majority of alloy steels show maximum tensile-strength values and minimum ductility in the range of 400 to 650 deg. fahr. (205 to 350 deg. cent.)." This condition might serve to partially offset the drawing effect.

Mr. Malcolm should be thanked for pointing out that bolts do not fail by "crystallization" but by fatigue through the application of alternate stresses. Failure by "crystallization" in alternately stressed parts is a misconception which metallurgists meet on all sides and it is to be hoped that Mr. Malcolm's statement will be of service in correcting this wrong idea. It is well to emphasize also the point in the fifth paragraph of Mr. Malcolm's discussion. It is too seldom appreciated that by using a heat-treated steel it is possible to increase elastic limit and tensile strength with scarcely any loss of ductility, as compared with that of untreated mild steel. Mr. Malcolm's reference to "dirty steel" is very timely. The presence of a large amount of inclusions in steel subjected to large alternating stresses is almost sure to cause trouble.

[1] Trans. A.S.M.E., vol. 46 (1924), p. 399.

No. 1981

THE VALUE OF HIGHER STEAM PRESSURES IN THE INDUSTRIAL PLANT

By William F. Ryan,[1] Syracuse, N. Y.
Member of the Society

This paper aims to point out the value of higher steam pressures in the industrial plant. It is shown that the gains from high pressure are greater than in the central station, and except for the question of suitable feedwater for high-pressure boilers, the problems involved are less difficult. The relative cost of power for varying initial pressure is estimated. The relative efficiency of turbine and engine prime movers, and the application of higher pressures to manufacturing equipment, are discussed. It is indicated that pressures up to the present commercial limit, about 1200 lb. per sq. in., may be used advantageously in the industrial plant which has a high load factor, and in which the demand for power is high, in proportion to the demand for process steam.

THE value of steam pressures higher than 400 lb. per sq. in. in the central station is a debatable subject; at least it is still being debated. There can be little question, however, that higher pressures can be used to advantage in industrial plants which use steam for manufacturing purposes.

2 The increase in thermal efficiency obtained by using pressures higher than 400 lb. per sq. in. in the condensing plant can hardly exceed 20 per cent. In the industrial plant the gain may be 75 per cent or greater, if the use of higher initial pressure permits the substitution of non-condensing prime movers, whose exhaust steam is used in process, for condensing units.

3 The application of higher pressures to large condensing units has been discussed at length in the Transactions of the Society. It is the purpose of this paper to initiate a discussion of the same problem as it affects the industrial plant.

4 The opportunity for generating cheap power, in the average industrial plant, is limited. If steam is used in process, or for

[1] Power Engineer, The Solvay Process Co.

Contributed by the Power Division and presented at the Annual Meeting, New York, November 30 to December 4, 1925, of The American Society of Mechanical Engineers.

heating, power may be generated very cheaply, but only to the extent that the exhaust from the prime movers can be utilized. Many plants which use no process steam are compelled to generate their own power on account of the high cost or low reliability of public-service current in some localities. It rarely happens that such plants have an unlimited supply of cold water for condensing purposes, and their ability to generate power at any reasonable cost is limited by the amount of steam that can be condensed with the available water. In either case, unless the power demand is low it is desirable to get the maximum practicable amount of power out of every pound of steam before it is condensed, either in manufacturing equipment or in condensers.

5 The necessity for this maximum of power extraction is, in most industries, increasing year by year. The installation of labor-saving machinery usually means the substitution of electrical or mechanical power for hand labor, and improvements in this direction are usually accompanied by an increase in power demand. On the other hand, improvements in manufacturing equipment are tending to reduce the consumption of steam per unit of production. As a result, many plants that have been able to generate all of their own power at very low cost, using the exhaust from prime movers in process, are finding that their power demand exceeds the condensing capacity of their steam-using equipment. Others are content to use obsolete and wasteful equipment because, if improvements were made, they would have an excess of exhaust steam from prime movers.

6 The economic considerations in applying high steam pressures to the industrial plant are fundamentally different from those of the central station, and the problem must be attacked in a different manner. Here we are not concerned with cycle efficiencies. Provided that the heat in the exhaust is fully utilized, the efficiency of the cycle is always 100 per cent. The actual efficiency is as close to 100 per cent as mechanical, electrical, and radiation losses will permit. This is true for any initial or final pressure and temperature. The water rate of the prime mover is as important in the industrial plant, as in the central station, not because a low water rate saves fuel, but because it permits the generation of more power from a given quantity of steam. Mechanical, electrical, and radiation losses are relatively more important than in the central station. A radiation loss which amounts to 10 per cent of the total heat consumption in a condensing unit would amount to more than 30 per cent in a high-back-pressure machine. Certain other losses, for instance leakage past turbine blades, while objectionable because they reduce the amount of power available from the fixed quantity of steam, are less vital than in the central station, because all heat which is not converted into useful work, or lost in radiation and friction, is delivered to the process in the exhaust steam.

Effect of High Initial Temperature

7 High initial temperature is essential to the highest steam economy. In considering how high a pressure should be used in any particular plant, it is almost axiomatic that the temperature should be as high as the materials of construction permit. Individual designers will differ as to what the practicable temperature limit is, but there is a fairly general agreement in this country that it lies between 650 and 750 deg. fahr. Drs. Mellanby and Kerr in one of the most interesting papers[1] on the subject published this year conclude that 900 deg. fahr. is an entirely feasible temperature, for either stationary or marine plants. Their arguments are not unconvincing from a metallurgical point of view, but they describe the range from 750 to 1100 deg. fahr. as "the field of danger and of progress." Into this field the industrial plant may well be content to let the central station blaze the trail. The central station has demonstrated that 750 deg. fahr. is an absolutely practicable temperature and that a properly designed plant may operate with an average temperature of at least 725 deg. fahr.

8 Fig. 1 shows how the gross output of a non-condensing turbine, per unit weight of steam used, is increased by increasing throttle temperatures. The gain is relatively less at the higher back pressures than at the lower back pressures, but is considerable under any condition. A turbine operating between 565 and 180 lb. per sq. in. absolute will deliver about 6 per cent more power per unit weight of steam with an initial temperature of 725 deg. fahr. than it will with an initial temperature of 650 deg. fahr.

9 A more important point than the increase in output per unit weight of steam is that high initial temperature produces a higher quality of exhaust steam. This makes it possible to use high initial pressures without resuperheating the steam during its expansion.

10 In all comparisons of pressures, a throttle temperature of 725 deg. fahr. has been assumed. It is felt that a plant can be designed for safe and dependable operation at that temperature, and the use of a materially lower temperature will not yield the best obtainable results from high pressures.

Effect of Initial Pressure

11 Fig. 2 shows the effect on the output of increasing throttle pressures per unit weight of steam, for non-condensing turbines. This is without resuperheating of the steam during its expansion and produces exhaust steam of varying quality, depending upon the initial pressure.

[1] The Limiting Possibilities in Steam Plants. A. L. Mellanby, D. Sc., and William Kerr, Ph. D., a paper read before the North-East Coast Institution of Shipbuilders, Newcastle-on-Tyne, February 27, 1925, and published in *Engineering*, London, March 6, 13, and 20, 1925.

12 It will be noted that the optimum pressure increases as the back pressure increases. With a final pressure of 1 in. Hg absolute there is no gain by increasing the throttle pressure above 750 lb. At 180 lb. per sq. in. absolute back pressure, the maximum is obtained with an initial pressure of about 1550 lb. per sq. in.

13 For comparison with previous studies, the output per unit weight of steam for a condensing turbine has been worked out.

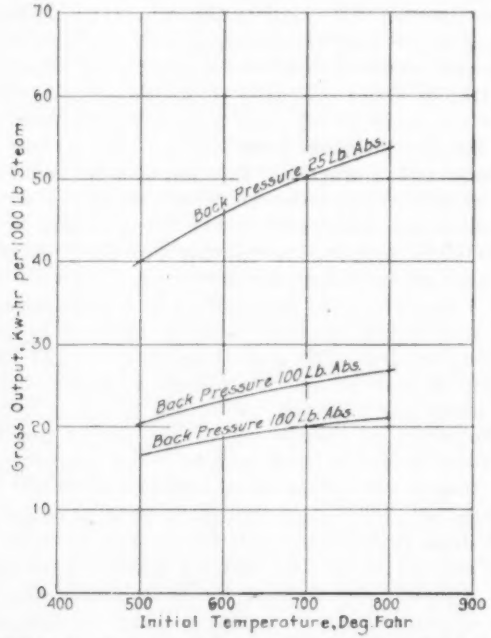

FIG. 1 EFFECT OF INITIAL TEMPERATURE ON POWER OBTAINABLE FROM STEAM AT 565 LB. PER SQ. IN. ABSOLUTE PRESSURE, FOR VARIOUS BACK PRESSURES

Fig. 2 compares the result of this calculation with that of Messrs. Hirshfeld and Ellenwood.[1] The estimated output per unit weight of steam at high vacuum is considerably less than that calculated in this earlier paper. The difference is due chiefly to different assumptions in regard to turbine efficiencies. Messrs. Hirshfeld and Ellenwood based their assumptions on 30,000-kw. units operating at the most economical load and under test conditions. This present paper is more concerned with units of 1500 to 5000 kw. capacity, operating under the average part-load conditions of

[1] High Pressure, Reheating, and Regenerating for Steam Power Plants, by C. F. Hirshfeld and F. O. Ellenwood, Trans. A.S.M.E. vol. 45, p. 663.

the industrial power plant. While there is a considerable difference in numerical values, the general characteristics of the curves are the same, although the maximum output is found at a slightly lower pressure and the decrease in output with increasing pressures is more rapid at the higher pressures. For the purpose of comparing one pressure with another, the two methods of calculation are in reasonably close agreement.

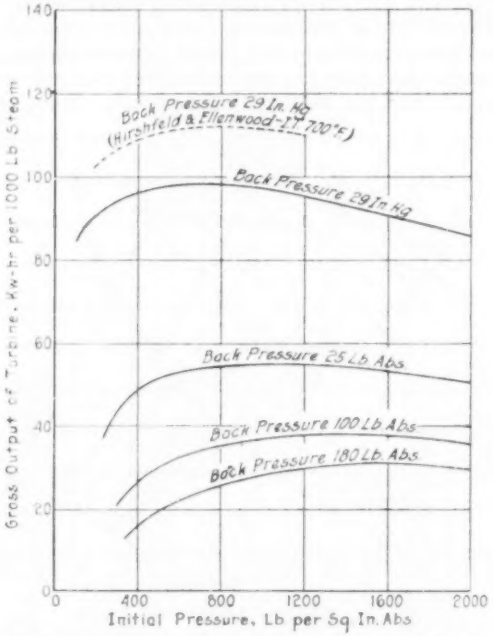

FIG. 2 EFFECT OF INITIAL PRESSURE ON POWER OBTAINABLE FROM STEAM AT 725 DEG. FAHR., FOR VARIOUS BACK PRESSURES

14 No estimate has been made of the benefits derived from high pressures in the condensing industrial plant operating with poor vacuum. The gain in this case, however, is twofold, economy being improved not only by the higher initial pressure, but also by the better vacuum obtainable when condensing smaller quantities of steam.

METHOD OF CALCULATION

15 The method of estimating the power outputs shown in Figs. 1 and 2 is as follows:

16 In accordance with facts brought out in the discussion of high pressures[1] at the annual meeting of this Society in 1923,

[1] Economy Characteristics of Stage Feed Heating by Extraction, E. H. Brown and M. K. Drewry, Trans. A.S.M.E., vol. 45, p. 713.

it is assumed that the expansion line of a steam turbine above or below the saturation point may be plotted as a straight line on the Mollier diagram without material error, and also that the stage efficiency of a turbine may be considered to be 10 per cent less in the wet zone than it is above the saturation temperature.

17 Stage efficiency of the turbine is also assumed to vary with the density of steam, in accordance with the equations developed by Mellanby and Kerr in the paper above mentioned.

18 As a standard of comparison, the average efficiency relative to the Rankine-cycle efficiency of several non-condensing machines of various ages and makes has been calculated from the steam consumption and from the quality of the exhaust. These efficiencies

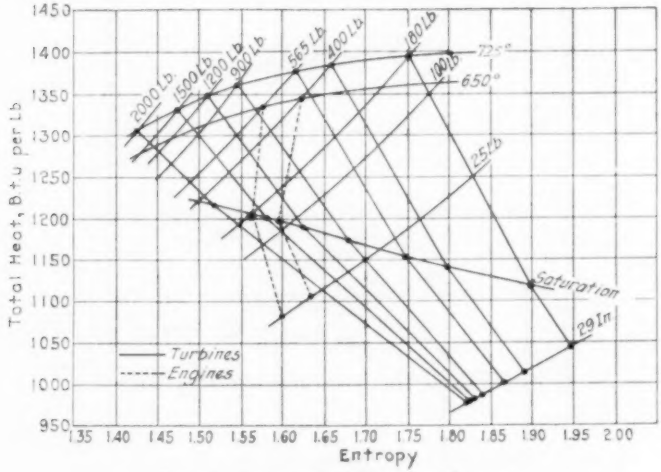

FIG. 3 ASSUMED EXPANSION LINES

have been compared with the original manufacturers' guarantees, and the ratio of average performance to guaranteed efficiencies has been applied to the machines of more recent and more efficient design.

19 The relative efficiency for different initial pressures is shown by the slope of the assumed expansion lines in Fig. 3.

20 There is competent authority[1] for the statement that high pressures do not decrease the efficiency of turbines, or even that efficiency increases with pressure. This statement is not in accord with recent guarantees on high-back-pressure machines for moderately high initial pressures (400 to 600 lb. per. sq. in.) and is contrary to the usually accepted theories in regard to flow of fluids.

[1] Hans Dahlstrand, A Review of Steam-Turbine Development, *Mechanical Engineering*, October, 1925, vol. 47, no. 10, p. 800.

21 The turbine efficiencies indicated in Fig. 3 are much lower than those assumed in previous papers on high pressures. The variation of efficiency with throttle pressure is substantially the same as that assumed by almost all of these studies except the paper by Hans Dahlstrand previously mentioned.

22 No operating data are available for small non-condensing machines operating at pressures higher than 400 lb. per sq. in. The expansion lines shown in Fig. 3 are in close agreement with actual

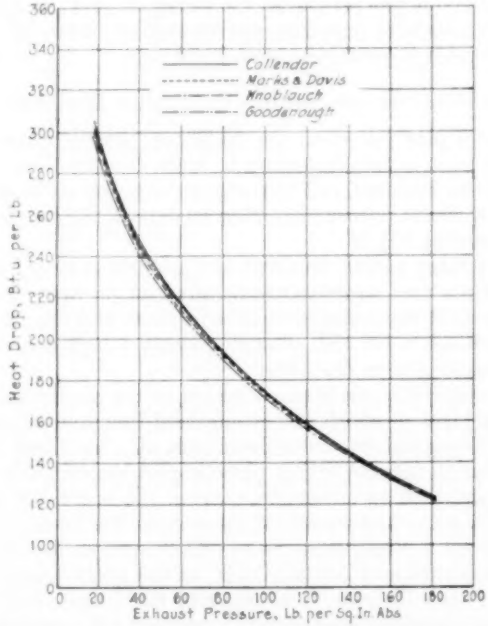

FIG. 4 ADIABATIC HEAT DROP FROM 565 LB. PER SQ. IN. ABS. AND 725 DEG. FAHR., COMPUTED FROM VARIOUS STEAM TABLES

performance of non-condensing units varying in size from 600 to 3500 kw., for initial pressures of 150 to 350 lb. per. sq. in.

23 There is very little agreement among manufacturers in their guarantees of steam consumption at the higher pressures. However, if the output of a turbine per unit of weight of steam is calculated from efficiencies assumed in Fig. 3, the result will lie somewhere between the half-load and full-load guarantees in every manufacturers' proposal that the author has had the privilege of seeing. While the assumed efficiencies may be materially in error, it is believed that the method of calculation provides a fair basis for comparing one initial pressure with another.

24 Callendar's steam tables were used for all the calculations in this paper, on account of their completeness and convenience of arrangement in the region of high pressures. The use of another steam table would make little difference in the comparison of results obtained from various pressures. Fig. 4 shows the adiabatic heat drop from 565 lb. per sq. in. absolute and 725 deg. fahr. as calculated from various tables. Callendar shows a slightly higher value than the others, especially for the lower back pressures. The agreement with Marks and Davis and with Goodenough is very close, within 1.5 per cent over the range of back pressures considered. At low back pressures, the discrepancy between Callendar and Knoblauch amounts to about 3 per cent.

Initial Pressure for Specific Quality of Exhaust Steam

25 For a plant in which the quality of process steam is of no consequence, it is only necessary to know the quantity and pressure of steam required, and the amount of power to be generated. The proper initial pressure can then be determined by the method used in plotting Fig. 2.

26 For many plants, however, the problem is more specific. A fairly definite final superheat is required, or, in many cases, it is preferable that there should be no superheat at all in the exhaust steam, although even with saturated steam a high degree of dryness is almost always desirable.

27 One specific case is the selection of the proper boiler pressure for a plant in which several hundred thousand dollars worth of pumps, compressors, turbo-generators, etc., have been installed, all designed to operate with a throttle pressure of 150 lb. per sq. in. gage, and with an initial superheat of 135 deg. fahr. To maintain this pressure and superheat throughout the works it is necessary to carry a steam pressure of 165 lb. per sq. in. gage and a steam temperature of 535 deg. fahr. at the boiler plant, which is located at some distance from many of the larger prime movers. The stated superheat is essential to the most efficient operation of prime movers throughout the works. It is of doubtful value in many processes, but no positive detriment from this relatively mild superheat has been encountered.

28 To obtain more power per unit of steam generated, the simplest procedure is to install higher-pressure boilers in the present boiler house, piped directly to high-pressure prime movers which exhaust into the existing live-steam mains. This makes a very simple installation. The only extra-high-pressure piping in the works is that from the boilers to the turbines and the feedwater piping to the new boilers. The exhaust is delivered at the most logical point for distribution to the process, and no existing equipment need be disturbed. It is necessary, in this case, to choose an initial pressure and temperature which will give not only the desired amount of power, but also a temperature of 535 deg. fahr. at the exhaust pressure.

29 Fig. 5 shows the quantity of power that can be generated per thousand pounds of steam used with an initial temperature of 725 deg. fahr., a final pressure of 180 lb. per sq. in. absolute, and a final temperature of 535 deg. fahr. The net power output is plotted as a function of the initial pressure. The method of calculation is the same as that employed in plotting Fig. 2, except that power for excitation and for boiler-feed-pump operation, which is a considerable item at the higher pressures, is deducted

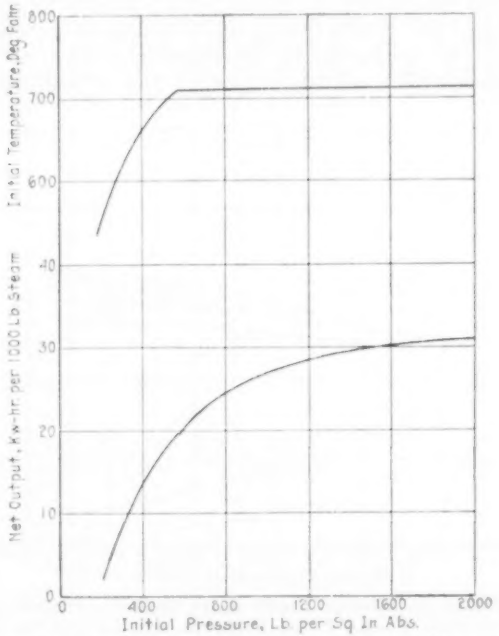

FIG. 5 INITIAL TEMPERATURE REQUIRED TO OBTAIN EXHAUST STEAM AT 180 LB. PER SQ. IN. ABS. AND 535 DEG. FAHR., WITH POWER DEVELOPED PER 1000 LB. OF STEAM. RESUPERHEATING USED AT PRESSURES ABOVE 565 LB. PER SQ. IN. ABS.; REHEAT PRESSURE, 400 LB. PER SQ. IN. ABS.

from the gross output. Excitation is taken at 1 per cent, the excess feed pressure at 10 per cent and the overall efficiency of the pump at 50 per cent.

30 For initial pressures up to 565 lb. per sq. in. absolute, the desired final conditions can be obtained without reheating. Above that pressure a resuperheater must be employed. In calculating the power output with reheating, a pressure drop of 5 lb. per sq. in. through the resuperheater has been assumed. This is the actual pressure drop obtained in one central station using the reheating cycle. With higher turbine efficiencies than those assumed in this

paper, the maximum initial pressure, without reheating, would be correspondingly lower.

31 For pressures requiring resuperheat, the reheat pressure is taken at 400 lb. per sq. in. Higher outputs would be obtained by using higher reheat pressures, as shown in Fig. 6, especially at the highest initial pressures.

32 For pressures up to 900 or 1000 lb. per sq. in. absolute, the loss resulting from reheating the steam after it leaves the turbine, rather than during the course of its expansion, is small. The greater simplicity of this design, which eliminates two steam connections to the turbine and an emergency stop valve, would probably outweigh the small loss of power. This arrangement would

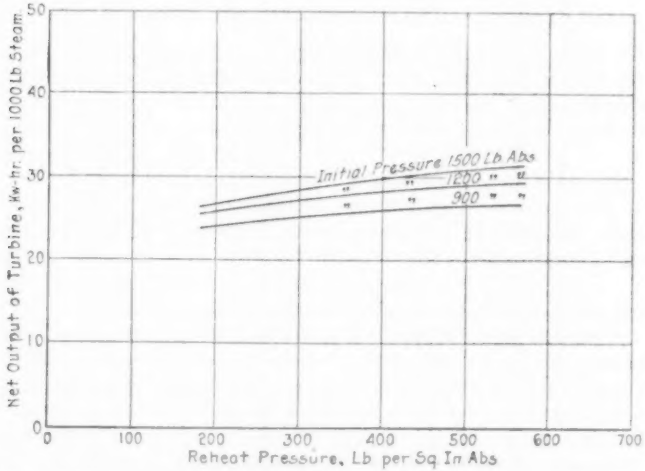

FIG. 6 EFFECT OF REHEAT PRESSURE ON OUTPUT. INITIAL TEMPERATURE, 725 DEG. FAHR.; FINAL PRESSURE, 180 LB. PER SQ. IN. ABS.; FINAL TEMPERATURE, 535 DEG. FAHR.

have the added advantage that steam could be piped without reheating to those processes in which superheat is of no value, if an arrangement could be made which would insure a regular supply of steam to the reheater.

33 In the industrial power plant, the elimination of the reheater is a material advantage. Some central-station designers have taken the operating dangers inherent in the resuperheater rather casually, and the safety devices are not elaborate. Others have provided rather complicated protective equipment, and still others have been deterred from its use altogether on account of the problems presented both in design and in operation. So far, no operating difficulties in actual use of the resuperheater have been reported, but the greater simplicity of using steam without

reheating is an advantage which should not be sacrificed without a very material gain in some other direction.

Cost of Plant and Equipment

34 Fig. 7 is an attempt to show the relative costs of boilers and boiler plants for various pressures. The cost of a boiler is, of course, a function of many things besides its pressure. The size, design, method of manufacture, and arrangement of heating surface all enter into the cost. The cost of the complete boiler plant is even more complicated, and actual costs will vary widely with location, the designer's skill, and the load factor for which the plant is designed. Fig. 7 shows at least the relative estimated

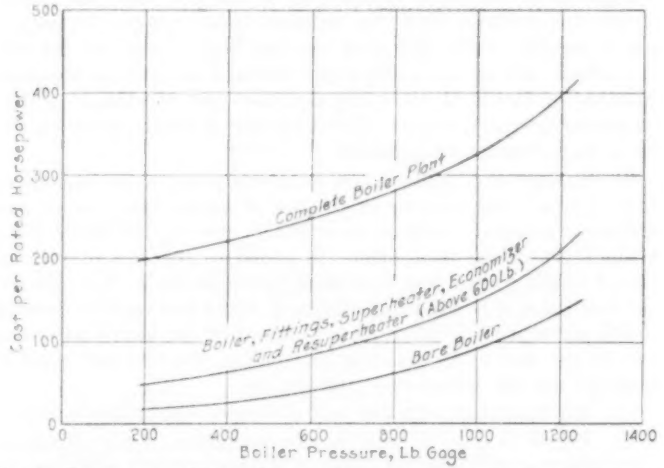

Fig. 7 Increase in Boiler-Plant Costs with Increase of Pressure

costs of a plant, for one particular set of conditions, for various pressures.

35 Actual cost would not, of course, be a smooth-curve function of the pressure, but would make definite increases at definite pressures.

36 The plant is designed to include sufficient economizer and air-preheating surface to give the same final flue-gas temperature (300 deg. fahr.) and as nearly as possible the same operating efficiency at all pressures. No charge is included for real estate. The cost of building is about 30 cents per cubic foot. Stoker firing is assumed and stokers and auxiliaries are designed for a maximum output of 350 per cent of boiler rating.

37 It will be noted that while the cost of a 1200-lb. per sq. in. boiler is about four times that of a 400-lb. unit, the cost of the complete plant at the higher pressure is less than twice as great.

38 The cost of a turbo-generator increases relatively less with increased pressure. While the prices vary widely with different designs, a machine for 1200 lb. per sq. in. initial pressure and 180 lb. per sq. in. back pressure will not cost more than 50 per cent more, installed, than a machine of the same size and back pressure for 400 lb. per sq. in. initial pressure.

Cost of Power

39 Cost of power has been estimated as follows: To the cost of boiler plant as estimated in Fig. 7 has been added the cost of turbo-generators of sufficient capacity to use the steam generated by the boilers when operating at 200 per cent of rating. The average boiler output is assumed to be 160 per cent, but higher ratings are necessary when one or more boilers are out for inspection or repairs. From this total cost has been subtracted the cost of a 200-lb. per sq. in. boiler plant designed to produce the same quantity of steam at the same pressure and superheat as that exhausted by the turbines. This difference is taken as the cost of the power-producing equipment.

40 Straight-line depreciation in fifteen years, taxes and insurance at 3 per cent, and the excess cost of maintenance, labor, and supplies over that required in a 200-lb. per sq. in. boiler plant without generating equipment are added to the cost of fuel required to supply the heat consumed in the turbines. For estimating fuel costs, a boiler-plant efficiency of 85 per cent is assumed at 100 per cent load factor, 80 per cent at 50 per cent load factor, and 72 per cent at 25 per cent load factor. Cost of coal is taken at $5 per net ton, delivered at the stokers.

41 The resulting costs per unit of output are shown in Fig. 8.

42 Fig. 8 is only a graphic representation of the estimated costs of power and does not indicate the best steam pressure to be used. If a similiar chart were prepared for a central station, the costs should include return on invested capital, and the optimum pressure would be that at which cost is a minimum. As previously stated, the problem in the industrial plant is to get the required power from the available steam. A chart like Fig. 8 will show how much of it may be obtained more cheaply from high-pressure turbines than from other available sources.

43 At one plant where a high-pressure installation is being contemplated, power cannot be generated condensing for less than one cent per kilowatt-hour, and power, dependable or otherwise, cannot be purchased for less than 0.75 cent per kilowatt-hour. The load factor is exceptionally high, averaging over 80 per cent for the year, and over 90 per cent for any month. If dependable equipment could be obtained, it would pay, in this case, to use pressures up to 1200 lb. per sq. in., if that were necessary, to generate the required amount of power. The return on the difference in

invested capital for a 1200-lb. per sq. in. non-condensing plant over a 200-lb. condensing plant would be at least 50 per cent.

44 The average steam demand at 165 lb. per sq. in. gage pressure and 535 deg. fahr. total temperature is over 750,000 lb. per hr. It would be possible in this plant to generate 15,000 kw. continuously at 565 lb. per sq. in. initial pressure or 20,000 kw. at 1000 lb. per sq. in. initial pressure, in turbines exhausting steam at the pressure and superheat now furnished by the boilers. With

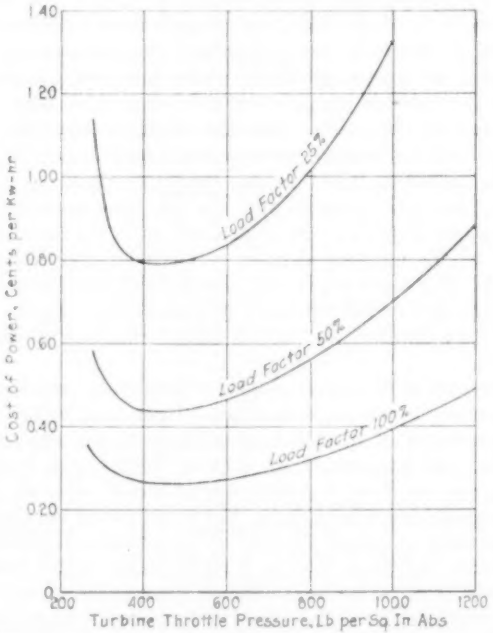

FIG. 8 EFFECT OF INITIAL PRESSURE ON COST OF POWER. THROTTLE-STEAM TEMPERATURE, 725 DEG. FAHR.; EXHAUST-STEAM PRESSURE, 180 LB. PER SQ. IN. ABS.; EXHAUST-STEAM TEMPERATURE, 535 DEG. FAHR.

higher turbine efficiencies than those assumed in this paper, the outputs would be correspondingly higher.

THE RECIPROCATING PRIME MOVER

45 For the central station with peak loads of 30,000 to 700,000 kw., the only practicable prime mover in the present state of the art is the turbine. For the industrial plant with its relatively low power demand, the reciprocating engine must be considered, at least in part of the field now under consideration. The mention of an engine-generator has a reactionary sound, but when its steam

economy in the range of pressures under consideration is compared with that of the high-back-pressure turbine, the claims of this type of prime mover cannot be altogether ignored. With a back pressure of 180 lb. per sq. in. absolute as much power can be generated by 1000 lb. of steam in an engine-generator operating with an initial pressure of 400 lb. per. sq. in. absolute and an initial temperature of 650 deg. fahr. as can be generated in a turbine operating at 500 lb. per sq. in. absolute and 725 deg. fahr. The superheat in the exhaust is of course less, and if the same final conditions are to be obtained, the exhaust must be resuperheated.

46 Fig. 9 compares the guaranteed performance of a steam turbine and an engine operating under identical conditions, except that the throttle temperature is 75 deg. fahr. higher for the turbine than for the engine. Manufacturers are unwilling to build engines, as yet, for temperatures higher than 650 deg. fahr.

47 The radiation losses in the engine are much greater than in the turbine, and consequently the net heat consumption per unit of output is greater. As shown in Fig. 9, the overall thermal efficiency of the engine-generator is relatively low, especially at light loads. For other types of prime movers, such as pumps and compressors, the overall efficiency of engine drive may be superior to that of the turbine, due to the higher efficiency of the driven apparatus.

48 The cost of an engine-generator to operate between 400 and 180 lb. per sq. in. absolute is about 30 per cent higher than for a turbo-generator of the same capacity operating under the same conditions. On account of its greater weight and larger space requirements, the total installation cost would be twice as much for the engine as for the turbine. For equal power developed from a given amount of steam, the cost of the engine installation would be somewhat less than that of a turbine due to the fact that a lower pressure could be used. Which type prime mover will furnish the cheaper power is largely a function of load factor, the higher first cost of the turbine plant being the more important factor at low load factor, and the higher heat consumption of the engine being the more important factor at high load factor.

49 No brief is held for the reciprocating engine for the generation of power. The greater space occupied, the very much greater weight, the contamination of the exhaust with lubricating oil, and problems presented by stuffing boxes at very high pressures, all go far to counterbalance the better steam economy. While a greater amount of power can be generated in the engine from a given quantity of steam, the net effect at the coal pile is the same as if part of the power were generated in high-back-pressure turbines, and the remainder in condensing units. A solution most to be desired is that the turbine designers may produce small-sized high-pressure machines which approximate the steam economy of the reciprocating engine, without greatly increasing the selling

price. Considerable progress has been made in this direction lately on small condensing units; similar development has been made in the high-pressure non-condensing field abroad, and equal progress may be expected soon in this country.

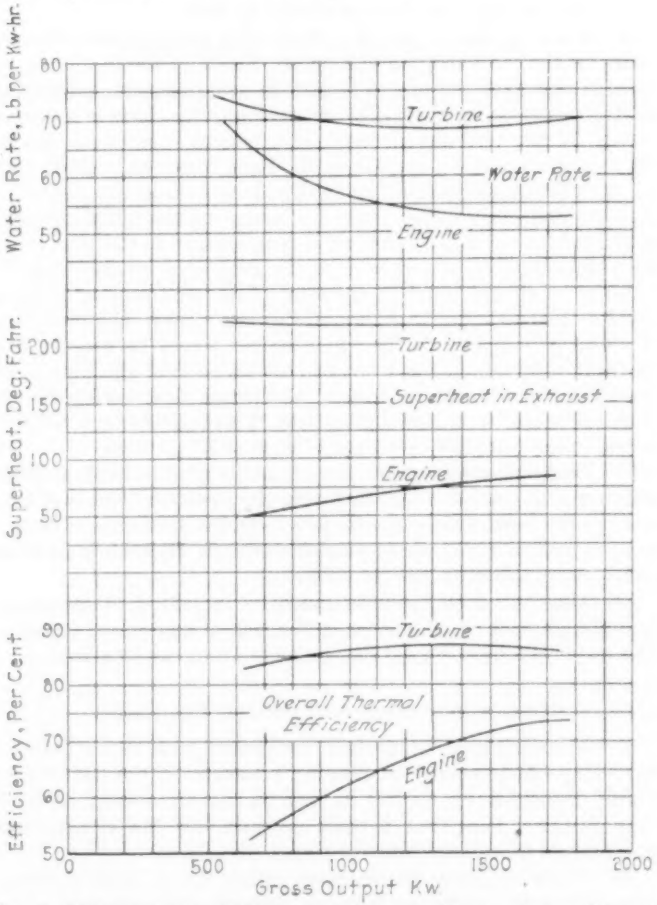

FIG. 9 COMPARISON OF ENGINE AND TURBINE PERFORMANCE. INITIAL PRESSURE, 400 LB. PER SQ. IN. ABS.; FINAL PRESSURE, 180 LB. PER SQ. IN. ABS.; INITIAL TEMPERATURE FOR ENGINE, 650 DEG. FAHR.; FOR TURBINE, 725 DEG. FAHR.

THE VALUE OF HIGHER BACK PRESSURES

50 In plants where the power load is low in proportion to the exhaust-steam demand, higher initial pressures may be desirable in order to permit the use of higher back pressures.

51 The capacity of many kinds of steam heating equipment is increased with increased pressure, or conversely, the size and cost of equipment for a given output is decreased with increased steam pressure. Higher pressures in evaporating processes usually increase the capacity, the heat economy, or both.

52 A typical case is that of a plant using multiple-effect evaporators for the concentration of a liquor. Steam at 150 lb. per sq. in. and 135 deg. fahr. superheat is used in prime movers, exhausting at 10 lb. per sq. in. gage. The exhaust steam, which is approximately 94 per cent dry, is used as a heating medium in triple-effect evaporators. By increasing the back pressure to 25 lb. per sq. in., a fourth effect could be used, reducing the steam consumption approximately 25 per cent. By increasing the initial pressure to 375 lb. per sq. in. and the initial temperature to 625 deg. fahr., the same quantity of power could be generated on this reduced quantity of steam. That is, the total power and heating demand could be taken care of with 25 per cent less steam. The total heat of steam as supplied to the prime mover would be greater, but the heat returned to the boilers in the drip from the first-effect evaporator would also be higher; the actual heat consumed per pound of steam would be slightly less. There should be a net fuel saving in the complete operation of at least 25 per cent.

53 Many plants are using live steam in process work at 40 to 100 lb. per sq. in. pressure. Reducing the pressure from that supplied at the boilers often results in a troublesome degree of superheat, as well as in a waste of available energy. High-back-pressure prime movers remedy this situation.

54 More advance has been made by industrial plants in the field of high back pressure than in the field of high initial pressure. The latter is a most logical development from the former.

Higher Pressures for Heating Equipment

55 Manufacturers of heating equipment such as evaporators have made little change in limiting pressures in the last twenty years. Pressures up to 100 lb. per sq. in. were used on large evaporators at least twenty years ago, and very few, if any, large evaporators have been built for higher pressures.

56 While temperature increases rather slowly with increased pressure, it is usually possible to increase the efficiency of an evaporating process by increasing the steam pressure. With a highly concentrated liquor which has such a high boiling temperature that only single-effect evaporation can be used with steam at 100 lb. per sq. in. pressure, two effects could be used with a pressure of 250 lb. per sq. in. without reducing the mean temperature difference between the steam and the liquor.

57 The design of an evaporator with a shell 16 or 20 ft. in diameter for very high pressures presents some problems, but

few which have not been solved in the Scotch marine boiler. The advantages of higher pressure for process, particularly in plants with a low power demand, are so great that considerable development in this direction may be expected in the near future.

Feedwater for High-Pressure Boilers

58 Obtaining suitable feedwater is one of the more serious problems which will confront the majority of industrial plants contemplating the use of high-pressure boilers.

59 The decreasing solubility of calcium sulphate with increasing temperature makes many treated waters which are fairly satisfactory at 200 lb. per sq. in. pressure almost unfit for use at 600 lb. per sq. in. Dilution by blow-down is more costly at high pressures. The consequences of a rupture, resulting from scale formations, are more to be feared at the higher pressures. If possible, only distilled water should be fed to high-pressure boilers.

60 The condensing plant, provided the condensers are tight, returns 90 per cent or more of the total feed to the boilers as practically pure distilled water. The quantity of make-up is so small that the raw water can be evaporated without much expense and it is possible to feed boilers with water almost free from solids. In the industrial plant, condensate may be returned from some apparatus, but with direct-contact heaters, such as distillers, vulcanizers, and digesters, no condensate can be recovered.

61 In such cases a pure water supply could be obtained by condensing the exhaust in single-effect evaporators. These evaporators, fed with raw water, supply vapor to the process, and the condensed exhaust steam is returned to the boilers. No heat loss, other than radiation, is involved, and there is a material advantage in the fact that leakage is necessarily outward, and if the evaporators are properly operated, there is no possibility of contaminating the condensate with raw water. There is, of course, a loss of energy involved because the prime movers must exhaust at a higher pressure than that required in the process; otherwise, there would be no temperature head available for the evaporation of raw water. If a high temperature difference is maintained in order to keep down the cost of the evaporating equipment, this loss of energy may be considerable.

62 Some distilled feedwater may be obtained from most direct-contact heaters without loss of energy by flashing the hot discharge liquor and using the vapor for multiple-effect evaporation of raw water. The amount of water that can be recovered in this way is small. Vulcanizer drip at 320 deg. fahr., flashed in four stages and used in quadruple-effect evaporators would not yield more that 25 per cent of its own weight in distillate.

63 The cheapest method of obtaining pure feed may be, in the majority of cases, to evaporate the total make-up, multiple-effect, with steam at fairly high pressure.

Commercial Limitations

64 The limiting pressure for present-day plant design is set, not by the calculated minimum cost, but by the ability of manufacturers to supply the required equipment.

65 At this writing the boiler and valve manufacturers are the only ones who appear entirely willing to supply equipment for pressures much above 600 lb. per sq. in. gage.

66 Boilers of practically any capacity may be obtained for pressures up to 1200 lb. per sq. in. Valves at least as large as six-inch are obtainable for the same pressure.

67 At least one turbine manufacturer is willing to quote on small, high-back-pressure turbines for initial pressures of 1000 to 1200 lb. per sq. in., but this company prefers to build its first machine of this kind in a larger size than could be used by the majority of industrial plants. Other turbine manufacturers are reluctant, for the present at least, to quote on any turbines for pressures higher than 600 lb. per sq. in. This is undoubtedly a temporary condition and one which will be changed as soon as the operating data on the one higher pressure unit so far built are available.

68 Engines are in use operating at 400 lb. per sq. in. gage, and at least one engine builder is willing to supply reciprocating equipment for pressures up to 600 lb. per sq. in. gage.

69 Boiler, turbine, and valve manufacturers accept temperatures up to 750 deg. fahr., but the engine builders set a limit 100 deg. fahr. lower. While the lubrication problem at high temperatures is a formidable one, the difficulties confronting the engine designer would appear to be less, on the whole, than those which have been overcome by the turbine designer. If a field is created for the high-pressure steam engine, there is little doubt that it will be soon adapted to higher temperatures as well as higher pressures.

70 Central stations have demonstrated the practicability of 600 lb. per sq. in. as a working pressure, and of 750 deg. fahr. as a working temperature. They will soon demonstrate whether or not 1200 lb. per sq. in. is equally practicable. This leaves very little development or experiment necessary for the industrial plant. Boilers, boiler auxiliaries, and valves would be the same in the industrial plant as in the central station. Turbine design should be simpler on account of the much lower range of temperatures in the high-back-pressure machine than in the condensing unit. Leakage and other handicaps to the smaller-sized unit are not so

serious in this type of machine, where all of the heat not used in actual work or lost in radiation is available for use in process.

71 When both the plant designer and the equipment manufacturer are thoroughly awake to the possibilities of higher steam pressures in the industries, commercial development in this field should be even more rapid than it has been in the central station. Because the gains are greater there, and the problems less difficult, it is probable that the eventual field for the highest steam pressures is in the industrial plant.

DISCUSSION

E. D. DICKINSON.[1] It is impossible to overestimate the value of studies such as the author has made. As is nearly always the case, possibilities of gain in economy must be shown from the theoretical standpoint before they are realized in actual practice. This is but natural. Sometimes a whole art has to be modified, new materials found, and new processes discovered before the full commercial advantages can be realized.

The question as to what is the best steam pressure to be used should be determined entirely by economic considerations. It is only recently that engineers in general have considered 350-400 lb. gage pressure and 725 deg. fahr. steam temperature as entirely commercial, and in order that this result be attained it has been necessary for manufacturers of equipment to develop, standardize, and have on the market a complete line of apparatus.

Referring to Pars. 4 and 5, it is just possible that in some industries the use of an extraction-type turbine might work out to good advantage. This type of turbine, of course, is somewhat of a compromise. However, in nearly all manufacturing, the relation of power requirements to heat demand is constantly varying. In such cases as these it is often found that the extraction-type turbine serves a most useful purpose in maintaining the most economical average heat balance.

Referring to Par. 6, the leakage past the turbine blades will result in lower efficiency and therefore higher water rate for a given steam flow. This is of particular importance where the electrical demand is great in proportion to the demand for process steam.

Referring to Par. 9, the gain from resuperheating is due in great measure to decreasing the wet region in the turbine. It will be apparent, therefore, that in turbines using steam for process, where the wet region is relatively small, the gain that may be secured by resuperheating can be relatively small. It is very doubtful, therefore, whether it will be economical to resort to resuperheating

[1] Designing Engineer, Turbine Engineering Dept., River Works, General Electric Co., West Lynn, Mass. Mem. A.S.M.E.

in connection with turbines where the exhaust is used for industrial purposes.

Referring to Fig. 2, the curves showing output are entirely reasonable. It must be remembered, however, that as yet there are no data available on turbines operating with the higher steam pressures. It is quite possible, therefore, that with the development of the art and improvements in design of apparatus, the curves indicating output may be somewhat different.

Referring to Par. 20, it is true that the thermal efficiency increases with the increase in pressure or superheat with constant back pressure, over the range of pressures that may be considered practical for some years to come. Also, with given initial and final pressures the turbine efficiency increases as the initial superheat is increased. However, for a given application with all conditions fixed, except the initial pressure, there is a decrease in turbine efficiency with increase in initial pressure. This will be more apparent when the term "turbine efficiency" is resolved into its components, output and losses. Since the output is constant and since the only variable, losses, increases as the initial pressure increases, the turbine efficiency must decrease.

Referring to Par. 29, this suggested method of increasing the capacity and efficiency of a plant by the installation of high-pressure boilers in conjunction with the existing lower-pressure boilers and the use of non-condensing high-back-pressure turbines between the two, has been shown to be economically sound.

HANS DAHLSTRAND.[1] While the paper deals principally with the use of high pressures in industrial power plants, it is clear that, so far as designing suitable equipment is concerned, the problem will be much the same, irrespective of the type of power plant. It is therefore of considerable value to designers and builders of power plants and machinery to have all vital questions fully discussed.

It is true, as the author states, that in industrial power plants the gain in thermal efficiency will be very much greater with increasing pressures than in power plants operating with condensing steam-turbine units only, provided, of course, that steam requirements for process work will balance the power requirements. The steam requirements for process work in some plants are greater than the power requirements, and it has been customary to consider the efficiency of the prime mover of secondary importance. But it should be realized that the power requirements are likely to increase faster than the steam requirements. For this reason prime movers of high efficiency, with the highest practicable steam pressure and temperature, should always be considered.

[1] Mechanical Engineer, Turbine Engineering Dept., Allis-Chalmers Manufacturing Co., Milwaukee, Wis. Mem. A.S.M.E.

The author has referred to the writer's paper reviewing steam-turbine development [1] as containing the statement that high pressures do not decrease the efficiency of turbines, or even that efficiency increases with pressure. While the above statements conform with views expressed in the above-mentioned paper, it should be fully qualified so that no incorrect conclusions will be drawn.

There practically are no complete economy tests published at appreciably higher steam pressures than those existing in modern power plants. Consequently it cannot be definitely stated that efficiencies will increase with increases in initial steam pressure, but, as mentioned in the writer's paper, from observations made so far there is every reason to believe that the efficiency of steam-turbine units will not decrease with higher pressures, provided the most economical velocity and velocity ratio is used in the design. It is of course understood that in making the above statement the comparison between efficiencies at various pressures should be made in the superheated region of the steam, as increasing moisture in the steam decreases the efficiency.

With reference to the author's remarks on the slight agreement among manufacturers in their guarantees of steam consumption at higher pressures, it should be noted that practically no development of steam turbines with high initial steam pressure and with various back pressures has been undertaken, for the reason that there has been no market for such units up to this time. Consequently the manufacturer has endeavored to use older developments as much as possible in estimating the propositions that have been considered. Another condition detrimental to an efficient design exists in that no standard pressures have been adopted for such units. The initial steam pressure varies from about 250 lb. per sq. in. gage to 800 lb. gage, with back pressures ranging from about 5 lb. per sq. in. gage to 150 lb. gage. It is obvious that the manufacturer is greatly handicapped in attempting to design and build efficient steam turbines of this type under such conditions with the limited market existing today.

It would seem that with the experience already gained from operation with high steam pressures in large power plants, a standard steam pressure could safely be adopted for industrial plants. This would greatly simplify the development of this type of steam-turbine unit, as well as render it less expensive and more economical to operate.

ALEX. M. ORMOND.[2] The author has stated that the use of high-pressure steam as outlined would be applicable only to plants in which the demand for power is high as compared to the demand

[1] A Review of Steam-Turbine Development. See p. 283.

[2] Chief Engineer, Savannah Sugar Refining Corpn., Savannah, Ga. Mem. A.S.M.E.

for process steam; also that a high load factor is necessary. This confines the use of high-pressure steam to very narrow limits. Most of the industrial plants of this country have boiler plants of less than 5000 hp. In a plant of this size, high-pressure steam would not be applicable on account of the small size of the units, both boilers and turbines.

A high load factor is essential for high-pressure steam. Very few industrial plants operate at high load factor. The production capacity of our industrial plants is far in excess of the demand for the products. A plant operating 250 days per year at 75 per cent of capacity would be a typical example. Then there are periods of depression in all industries when plants shut down entirely, which in the case of the high-pressure plant would mean a larger investment lying idle.

Most plants have a shut-down period each year besides Sundays and holidays. At such times light and some power is required, but no process steam. These items would have to be carefully considered before reaching a decision as to whether a high-pressure plant would be more economical.

The fluctuating demand for power and process steam also militates against the high-pressure plants. A survey of a plant might show that the demand for process steam exceeds the amount of steam produced in generating the necessary power. On account, however, of the fluctuating demand for process steam from hour to hour, or even from minute to minute, there will be times when it is necessary to bleed live steam into the exhaust mains, and at other times there will be an excess of exhaust which must be wasted. Under the above conditions the bleeder-type turbine operated in connection with a straight back-pressure turbine makes an ideal installation.

The high-pressure installation would require an economizer or air heater to reduce the temperature of the flue gases as outlined by the author. The evaporating equipment for supplying the make-up water should also be charged up against the high-pressure plant, and also the more careful supervision and higher grade attendants needed by it.

Charles H. Bigelow.[1] In using exhaust from a turbine for process steam, due consideration must be given to the fact that the demands for process steam do not coincide with the electric load on the turbine. The result is that either there is more demand for process steam than the exhaust will supply and the deficiency must be made up by steam direct from the boiler through a reducing valve, or else there is too much exhaust and the excess must escape to the atmosphere, so that one of these two valves is in

[1] Plant Engineer, Spicer Mfg. Corpn., South Plainfield, N. J. Mem. A.S.M.E.

operation practically all the time. The writer believes that a better net economy can be obtained from a bleeding turbine than from a non-condensing turbine, particularly as from 10 to 14 lb. pressure is needed, and that makes a high back pressure for the turbine to operate against.

The manner in which a power plant is operated has a great deal to do with the economy, and there are many cases on record in which the operating cost has been reduced 50 per cent or more, not by radical changes in equipment but by putting an intelligent, educated engineer in charge.

R. R. JONES.[1] The question of the utilization of power from higher steam pressures in industrial plants has another serious phase which we have not covered. Most modern plants are pretty well scattered. The one the writer has in mind distributes steam at a pressure of 175 lb. per sq. in. a distance of 3000 ft. Perhaps that is exceptional, but at the same time the distance to which the process steam must be distributed will be large.

In the event of using back-pressure turbines in a power plant or in a system of that kind, the engineer is faced with a difficulty. Shall he install the turbines and high-pressure boiler and distributing systems in his power plant, which should be more or less central, and then install large distributing mains to carry this lower-pressure steam to his process? Or shall he install his turbines or his other pressure-reducing devices in outlying sections of the plant, distributing high-pressure steam? The difficulties of either case will be considerable.

As more or less of a compromise, there can be obtained a considerable saving from a medium-pressure steam system, involving ordinary methods of distribution. A simpler kind of turbo-generator operating without attendance, perhaps, then could be used. This might solve the problem, not to the last degree of efficiency, but it would save a very considerable amount of money.

DAVID MOFFAT MYERS.[2] At one of the large rubber plants there is installed a 5000-kw. turbine-generator unit, taking steam at about 250 lb. per sq. in. and exhausting into the rubber vulcanizers at 80 lb. per sq. in. pressure. These vulcanizers were previously fed with direct steam from the boilers. Considering all charges — overhead, maintenance, depreciation, interest, everything, including fuel — on this unit, and charging or crediting the value of the exhaust steam delivered to the vulcanizers formerly fed by live steam, it delivers one kilowatt-hour for a little under half a cent.

[1] Chief Engineer, Firestone Tire and Rubber Co., Akron, Ohio. Mem. A.S.M.E.

[2] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.

J. R. McDERMET.[1] The author has presented a comprehensive discussion of his subject, applicable in the main to the chemical plant and the paper-manufacturing plant, and perhaps, also, to the textile-finishing plant. In the bulk of other manufacturing plants, two other facts, largely questions of adequate return upon invested capital, must be included, viz., that the plant operates, perhaps, only ten hours out of a twenty-four hour period, and that the cost of power averages generally less than two per cent of the cost of the finished product. The author's analysis for this type of plant, while technically exact, needs to be interpreted with business perspective, likewise.

The use of high pressure has not increased disproportionately the cost of the essential power-plant equipment. High-pressure minor equipment, such as small valves, water columns, etc., however, is prodigiously expensive compared with the lower-pressure article, and the complication of an industrial plant, from the standpoint of these smaller accessories, is ordinarily greater than for the central station. For such reasons it is improbable that an analysis from the central station cost can be applied directly to the power station of an industrial plant.

In the matter of boiler feedwater, the author has indicated the realization of several pertinent facts. It will be impracticable to chemically treat boiler water with precipitating reagents to reduce the calcium sulphate content to a point where it will not precipitate in high-pressure boilers, even with large concentrations of sodium carbonate to force chemical equilibrium. With zeolite softeners, the addition and substitution of a sodium ion is inevitable. Sodium carbonate is dangerous in high-pressure boilers, because its hydrolyzation under boiler operating conditions is a function of increasing temperature, and, hence, of increasing pressure.

The solubility of residual calcium sulphate left in boiler water is likewise a function of temperature, the solubility decreasing with temperature and reaching a minimum where the anhydrous salt exists as the stable phase. It is also significant that the precipitation of calcium sulphate occurs at the point where the water attains the maximum temperature, and that the precipitated material is generally not carried on by the water, but is deposited immediately at the point where precipitation occurs. In a boiler, the hottest part of the circuit is in the tubes directly over the fire, and this, likewise, is the most dangerous part of the boiler, from the standpoint of safety.

It is also an interesting fact that the laws of viscosity of water, as regulating bubble formation, indicate that the circulation in a high-pressure, high-temperature boiler is more uniform and more quiet, with less possibility of vapor binding of heating surface, the

[1] Research Engineer, Elliott Co., Jeannette, Pa. Mem. A.S.M.E.

higher the boiler pressure and temperature. Accordingly, less need exists for high velocities of circulation and large steam-liberating areas than with the conventional design. It is assured that circulation rates will be lower; accordingly, high-pressure boilers, operating at the same rate of evaporation, have relatively greater safety with the same factor of safety for materials than the lower-pressure boilers. The danger, however, as the author correctly asserts, is in the uncertainty of strength of materials, and the quieter circulation of the very-high-pressure boiler, while desirable with pure water, is increasingly dangerous with water containing calcium sulphate. The tendency of the calcium sulphate to precipitate at the point of maximum temperature is not disturbed by the boiler circulation or ebullition to the extent that would occur in the lower-pressure unit. A phenomenon, therefore, which is advantageous with pure water becomes an added source of risk with water containing calcium sulphate.

The writer believes it impractical, from thermal and investment considerations, to contemplate the evaporation and distillation of an entire boiler feed supply. The maximum limit is probably in the neighborhood of 25 per cent.

The high-pressure turbine, or in fact, the highly efficient moderate-pressure small turbine, will be available when there is a market for it.

R. E. HALL.[1] In his discussion of feedwater treatment for the high operating pressures he proposes, the author has clearly stated the unique relation of calcium sulphate to the formation of scale, and the increasing difficulties it engenders at higher pressures. With an increasing discharge of industrial waste liquors from year to year the concentration of soluble components, and especially of calcium sulphate, in the waters of lakes and rivers is certain to be greater. This condition requires more careful control of the boiler water for all operating pressures, especially in view of the high rates of driving characteristic today, and the economic necessity for reducing blow-down to a minimum. The resultant larger concentration of soluble components, especially sulphate, and of suspended insoluble material in the boiling water, renders the problem of preventing scale formation on the evaporating surfaces, and the growth of deposits in superheaters and on turbine blading, the more acute.

The criterion for preventing scale formation, regardless of pressure, is to establish such conditions in the boiler water that the adherent scale-forming substances, calcium sulphate and calcium and magnesium silicate, cannot be deposited therefrom on the evaporating surfaces. At lower pressures and rates of driving, and especially where the sulphate concentration is not high, this can be

[1] Bureau of Mines Experiment Station, Pittsburgh, Pa.

accomplished by the use of soda ash and lime, or by soda ash, or by a base-exchange process as the water may require. For such results the control of the treatment must be based upon tests of the boiler water itself.

At pressures such as those discussed in this paper, it becomes impracticable to establish the criterion defined above by the current practice of soda ash and lime, or by any process which does not carefully define the boiler water concentrations of treating chemical in terms of the chemical composition of the boiler water at the evaporating surfaces and the operating pressure. This follows because soda ash decomposes readily, even at 125 lb. gage pressure or lower, into caustic soda and carbon dioxide; and because the solubility of calcium sulphate is a function of the total sulphate concentration in the boiler water, and of the operating pressure.

Without the definite control indicated, even disproportionately large blow-down may fail in being effective. The author has noted the increased loss over lower-pressure operation of attempting to control concentrations by blow-down, but has not commented upon the difficulties connected with blowing down a boiler operating at the pressures he is discussing.

A material entirely stable at the pressures under consideration is sodium phosphate. A boiler water containing the necessary concentration of this treating chemical cannot deposit adherent scale, either sulphate or silicate, on the evaporating surfaces. In fact, its use at lower pressures would be highly desirable, were its cost not substantially greater than that of soda ash.

In condensing plants, therefore, in which the small percentage of make-up water can be readily evaporated, and the objective of treatment is to guard against corrosion and scale trouble arising from condenser leakage, the answer is definite for the high pressures under discussion. In fact, as far as these factors are concerned, the necessity for evaporating so small a quantity of make-up water is not apparent.

The author has discussed the possibilities of obtaining distilled water at low cost by utilizing the waste heat from various possible points in the system. He concludes that the prospects are not good for obtaining a large share of the make-up water in this manner. They need not be. Two answers are given:

a A primary treatment with lime and soda ash, or with a base-exchange process, followed by the maintenance of the necessary phosphate concentration in the boiler water, or a direct treatment with phosphate alone, would preclude scale formation. If any trouble should be experienced from deposits in the superheater and on the turbine blades or in the resuperheater, because of the greater concentration of soluble and suspended material in any moisture entrained in the outgoing steam, mechanical means are available for its separation.

b A second possibility lies in the use of a primary treatment with barium carbonate, followed by the maintenance of the necessary phosphate concentration in the boiler water. In the case of a feedwater high in sulphate, the total concentration of solubles would be very much reduced; in a chloride water, however, the value of the barium carbonate treatment is limited to the amount of sulphate therein.

One fact stands out clearly for the higher operating pressures: when continuity of operation demands positive inhibition of scale formation, the conditioning of the boiler water must be done with a treating chemical such as sodium phosphate, which is stable and effective at the operating pressure; and the maintenance of correct conditions must be judged by tests on the boiler water itself.

D. S. JACOBUS.[1] The necessity has been brought out of providing clean feedwater for high-pressure boilers. This is important and in line with the advancement that has been made in power-plant operation, the tendency being to use cleaner feedwater for all boilers irrespective of the pressure.

The effect of temperature on the strength of the steel in boilers is much less than in oil-cracking stills where the problem of providing a proper design has been met for temperatures of 800 deg. fahr. and higher. In a boiler the tensile strength of the steel subjected to the temperature of the saturated steam is about the same as at ordinary atmospheric temperatures. The superheater tubes are usually subjected to a temperature at which there is some falling off of strength, but the smallness of the tubes makes it easy to provide a proper factor of safety. In general it may be said that there need be no fear for the safety of a properly designed boiler to be operated at high pressures.

The pioneers in the use of high pressures for power-plant service deserve every credit for their enterprise in exploring a new field. It was felt that there would be troubles of one sort or another that could not be foreseen. The 600-lb.-pressure units and the controls made necessary through the use of interstage reheaters have operated in a most satisfactory way. We have yet to learn what will occur with the 1200-lb. units, but we have no fear of any trouble with the boilers and superheaters.

JOHN F. GLENN.[2] The problem of the industrial-plant engineer seems to be summed up in Pars. 25 and 26, and in the several paragraphs following Par. 39. These, together with Fig. 8, show the costs to be expected for plants of different steam pressures.

[1] Advisory Engineer, Babcock & Wilcox Co., New York, N. Y. Past-President A.S.M.E.

[2] New York Sales Manager, Edge Moor Iron Company, New York, N. Y. Assoc. A.S.M.E.

The problem of the boiler manufacturers for higher pressures will be overcome when the boiler has been brought to the pressure to which it is designed, but there yet remains to be proved the dependability and reliability of boiler fittings and accessories at these high pressures. It is to be expected that there will be numerous experiments and changes on various types of valves and other equipment having moving parts. The problem of application of these high pressures becomes a commercial one, i.e., cost of investment versus economy of operation, and is therefore individual for each type of industrial plant.

GUY B. RANDALL.[1] It is unquestionably true that not only high steam pressures, but other improvements, now under way, will prove more valuable to the industrial plant than to the central station.

Two or three points in the paper warrant special comment. In Par. 6, the author states that the industrial plant is not concerned

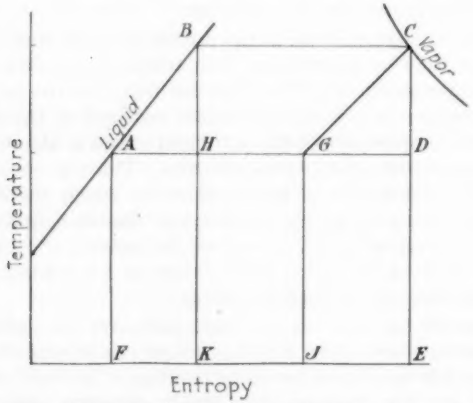

FIG. 10 GENERAL FORM OF TEMPERATURE-ENTROPY DIAGRAM FOR STEAM WITH LIQUID LINE DRAWN STRAIGHT AND REGENERATIVE LINE, *GC*, PARALLEL TO IT FOR SIMPLICITY OF CONSTRUCTION. NOT TO SCALE

with cycle efficiencies; that provided the heat in the exhaust is fully utilized the efficiency is always 100 per cent. Further on, he states that the water rate of the prime mover is as important in the industrial plant as in the central station, not because a low water rate saves fuel, but because it permits the generation of more power from a given quantity of steam. These two statements are rather incompatible. For, given the temperature and the amount of heat required for process work and, given a certain

[1] Chief Engineer, Power Station, The National Cash Register Co., Dayton, Ohio. Mem. A.S.M.E.

initial temperature and pressure of steam generation, if the maximum amount of mechanical effect is desired for a given amount of process heat, we are concerned with cycle efficiencies. Both the central station and the usual industrial plant are interested in getting the maximum mechanical effect per unit of heat rejected from the prime mover. The difference between them is this: The central station has to get all its power on a condensing basis, while the industrial plant using process steam throws heat into the sewer only when the power demand proves excessive for the process heat. The temperature-entropy diagram, Fig. 10, will make this clear. For geometrical simplicity the liquid line has been drawn straight.

Referring to the diagram

$$\text{Rankine efficiency} = \frac{ABCD}{ABCD+ADEF}$$

The mechanical effect per unit of heat rejected is $\frac{ABCD}{ADEF}$

On the other hand, with such a process as the complete regenerative cycle, the result is

$$\text{Regenerative efficiency} = \frac{ABCG}{ABCG+AGJF} = \frac{HBCD}{HBCD+HDEK}$$
$$= \text{Carnot efficiency.}$$

Whence the mechanical effect per unit of heat rejected at a given temperature is

$$\frac{ABCG}{AGJF} = \frac{HBCD}{HDEK}$$

$$\text{By inspection, } \frac{HBCD}{HDEK} > \frac{ABCD}{ADEF}.$$

Hence, the industrial plant designer is keenly concerned with the various cycle efficiencies, because his problems have greater variety, greater complexity, and therefore greater possibilities for individual treatment. The fact that the heat in the exhaust is fully utilized, i.e., that the thermal efficiency is 100 per cent, does not always tell the story. In fact, such a statement for the general type of plant under consideration is platitudinous unless we know that there is exact balance between power and heat demands. It is necessary to keep in mind that if the total demand for heat exceeds the by-product heat from power production, the thermal efficiency may still be 100 per cent and the heat cycle, both in type and in range, anything but what it should be. That is, the cycle may be too elaborate and too expensive.

In the same paragraph the author states that mechanical, electrical, and radiation losses are more important than in the central

station. This is by no means generally true, for the answer in any given case will depend on a number of considerations. For example, let us take two stations, one condensing at 28 in. vacuum and the other non-condensing at 28 in. vacuum. These stations are duplicates physically and thermodynamically except that the condenser is removed from the non-condensing plant and an industrial process-steam user substituted. The limiting Carnot thermal efficiency for the condensing plant with steam as the sole working substance is about 50 per cent. The limiting efficiency for the power and process-steam plant is 100 per cent. It is obvious that any heat loss up to the turbine throttle will cost the same in either plant. At the turbine exhaust a given heat loss will cost nothing in the condensing plant and 50 per cent of the throttle price in the non-condensing plant. So in these two cases all the heat losses up to the turbine throttle are of substantially the same importance, and these constitute the great bulk of the losses. The unit value of the heat losses from the turbine-generator will average 75 per cent of the throttle value for the non-condensing unit and 50 per cent for the condensing unit. While with increasing back pressure the value of the radiated losses of the turbine-generator alone are increased in the case of this ideal industrial plant, the bulk of the losses are of the same order of importance; and if there is a margin of live steam usage they are actually of less importance in the industrial plant than in the central station.

In Par. 59, the author refers to dangers of operation at higher pressures with other than pure feedwater. This is a point on which it would be interesting to have the actual experience of those who are operating boilers at high pressures, because however good the usual feedwater supply, eventually there are sure to be instances where scale, or the equivalent, causes tube failures. The writer is loath to believe that a hard water which has been properly treated, so that it causes no scale at 165 lb. pressure, will cause scale at 465 lb. pressure.

Mr. McDermet, in effect, stated that calcium sulphate is precipitated on the most active surfaces of the boiler and that this effect will be accelerated with increasing steam pressure and temperature. The writer is at a loss to understand this point of view. The tubes facing the fire are by all odds the most active tubes in a Stirling type of boiler, yet every one knows they are the ones least subject to scale. With a properly softened water they are, in actual practice, not subject to scale at all, unless they have previously been made rough on the inside due to improper water condition and consequent corrosion. But even in this case the deposit, with strict regard for texture, cannot be called scale. It is very easy to say that pure water is needed for high-pressure operation. But is it? Will not a properly treated " raw water " produce satisfactory operating results? The writer has heard of one plant

where 600-lb. steam pressure is used and where the operators were somewhat anxiously waiting for "something to happen." Finally a tube "let go," but no one knew it until the water tender found difficulty in keeping water in the boiler. It is perfectly right to be cautious and fearful in dealing with tremendous forces. But caution founded on fact is much more useful than that based on guesswork or superstition. The writer is not at all convinced that 400 or 600 lb. steam pressure will be found to be so much of an advance as 200 lb. pressure was 10 or 15 years ago, and certainly "pure water" is not essential for successful 200-lb. operation.

Germane to this discussion is the matter of industrial ovens. Electric ovens are popular today, but the advantages of the steam oven, when properly realized, so greatly outweigh electric ovens that there is no comparison for temperatures up to 450 deg. fahr. In connection with the use of higher steam pressures, such oven temperatures are readily obtained, but it is interesting to note that in the industrial plant it is thoroughly practicable to obtain local high pressures for an oven, for instance, by simply reversing the usual Rankine cycle and installing a motor and a steam compressor which is analogous to a simple motor-driven air compressor. The economic advantages of such a layout are very great compared to those of the usual electric oven.

For example, for an electric oven of 8-ft. cubical dimensions in which the temperature is to be maintained at from 350 to 450 deg. fahr., temperature elements operating at 1000 deg. fahr. will be used and air admitted through the resistor to the oven at 70 deg. fahr. In the same connection will be installed a high-grade thermostatic control, which will operate at 3 deg. temperature difference. Investigation has shown that such an installation frequently has temperatures varying 100 deg. fahr. in its different parts, while by simply making a frame of 1½-in. tubes and completely surrounding the oven with steam, an oven is obtained which will give a temperature of 350 or 400 deg. with a variation of temperature in the oven of the order of 10 deg.

The author mentions that dilution of boiler water by the blow-down process is more expensive at higher pressures. There are many more serious things than blow-down loss, and, in a practical way, altogether too much zeal and not enough judgment has been displayed in reducing it. In far too many plants the convenient practice of a blow-down of, say, two gages of water per boiler per watch is religiously followed, when two or three times as much would be economical. In an industrial plant which came under the writer's supervision the blow-down was increased from 2 per cent of the gross evaporation to approximately 8 per cent, the boiler water titrated daily, and a proper water softener and multiple baffle steam purifiers installed. Extra blows were given whenever the purifier traps showed signs of distress. The net effect of this was that steam of the order of purity of laboratory distilled

water was consistently obtained, the annual maintenance cost of the plant fell from $50,000 to $20,000, and the coal consumption from 36,000 tons to 24,000 tons. After all, what we are after is not the last two or three points of boiler efficiency, but sustained low total cost for power. One of the surest ways to get it seems to be the circumspect boosting of steam pressures in the manner so clearly set forth by the author.

F. O. Ellenwood.[1] In Fig. 3 there is shown on the total heat-entropy diagram the slope of the expansion line, and it is stated by the author in Par. 19, that this is a measure of efficiency of the unit. It has already been stated that for the reciprocating engine the line slopes to the left of the constant-entropy process. That can be due only to one cause, not cylinder condensation or initial condensation, but due wholly to radiation and condenser losses from the reciprocating engine. The reason for making the foregoing statement is that whatever loss occurs in the reciprocating engine, due to cylinder condensation, reappears in the heat content of the exhaust steam. These lines in Fig. 3 are supposed to represent the condition of the steam after it has been exhausted from the engine or the turbine, and not the condition of the steam during the expansion in the reciprocating engine.

The writer, therefore, has roughly calculated the radiation in the case of this reciprocating engine, in order to check out this dotted line for such an engine. It amounts to about 20 per cent of the delivered energy of such an engine. If that be the case, it would seem that there is an opportunity for considerable improvement on the part of the builders of such a reciprocating engine, because it seems doubtful that there is sufficient reason why the losses due to radiation from a high-pressure cylinder should be so large. The high-temperature steam will tend to cause higher radiation losses, but the specific volume of the high-pressure steam is also reduced, hence, the total loss due to radiation need not be appreciably larger than in the low-pressure engine. The writer, therefore, would suggest to the manufacturers of these high-pressure reciprocating engines the careful consideration by their designers of this question, assuming that Fig. 3 represents the facts as they exist today.

In Fig. 9 there appears to be a mistake in the curves representing the efficiency of the turbine and the engine at the bottom part of the diagram, in the upper part of which the steam rate of the turbine is very much larger than that of the engine. To make the diagram consistent it seems to the writer that it would be necessary to reverse the names "engine" and "turbine" in the lower part of the figure. In further proof of this statement, the author in

[1] Professor of Heat Power Engineering, Cornell University, Ithaca, N. Y. Mem. A.S.M.E.

Par. 45 says: "With a back pressure of 180 lb. per sq. in. absolute, as much power can be generated by 1000 lb. of steam in an engine-generator operating with an initial pressure of 400 lb. per sq. in. absolute and an initial temperature of 650 deg. fahr. as can be generated in a turbine operating at 500 lb. per sq. in. absolute and 725 deg. fahr. If the names "engine" and "turbine" be reversed in the lower part of Fig. 9 it will be consistent with Par. 45.

The paper brings out two facts of prime importance regarding high-pressure engines and turbines. There is an opportunity for the turbine manufacturers to build low-capacity turbines for use in industrial power plants with a higher engine efficiency than those now offered, and there is an opportunity for the reciprocating engine builder to compete by building a high-speed, low-cost, and low-radiation type of reciprocating engine. The paper also shows in a clear manner how important it is to consider the advantages to be gained by using high-pressure steam in the industrial plant.

HAROLD ANDERSON.[1] In Par. 7 the author makes the statement that it is almost axiomatic that the temperature should be as high as the materials of construction permit. It is doubtful, however, if full advantage should be taken of the highest possible temperature, as the average industrial plant does not have the careful supervision necessary for successful operation at these higher limits.

When determining the proper steam conditions, the initial temperature is generally not as important as the final temperature. Most industrial plants using process steam require a rather constant temperature, which is hard to control with superheated steam. For instance, in the pulp and paper industry, which ranks fifth among the major manufacturing groups in the United States, it is practically impossible to find a paper-mill superintendent who is willing to use superheated steam in the paper driers. The pulp mill also requires an easily controlled temperature for the digesters. The rational method to pursue, therefore, is to start with the final conditions desired in the exhaust steam and work back to the initial conditions, the initial pressure being determined from the amount of power required.

The author's statement in regard to the desirability of not using resuperheaters in the industrial plant is true as a general statement, although some exceptions may be found in individual cases.

There is nothing mentioned in regard to governing. As the process steam demand and the power demand in the industrial plant unfortunately seldom balance, it is in most cases necessary to add a condensing section to take care of the periods of fluctuating conditions. In some cases an accumulator may replace the condensing installation. The question of governing will influence both the cost of power and the operation.

[1] General Engineer, Westinghouse Elec. & Mfg. Co., South Philadelphia Works, Philadelphia, Pa. Assoc-Mem. A.S.M.E.

Fig. 7 is a curve giving the cost of boiler plants at increasing pressures. It is rightly pointed out that these curves should be broken, although they have been shown as smooth curves. Estimates made by the writer on high-pressure installations agree well with the author's curves, although the author's cost of the complete boiler plant is higher. It is possible, however, that the author's figures are confined to smaller plants, which would of course increase the unit cost.

The maximum boiler efficiency of 85 per cent is probably too high for all year running and cannot include any banking losses. If the efficiencies were reduced about 5 per cent, a closer estimate of the actual steam cost would be obtained.

The scheme of using the old low-pressure boiler equipment in conjunction with new high-pressure equipment will probably be employed in a large number of industrial plants. An installation of this kind has been mentioned on page 92 in the German paper *Hochdruckdampf*, published by the Verein Deutscher Ingenieure, in 1924. Installations of this kind must be carefully analyzed in regard to cost of conversion and the resultant fixed charges.

Fig. 3 shows the expansion lines for the various prime movers under consideration. It would be interesting to know how the two expansion lines for the engines were obtained, as they show a decrease in entropy above the saturation line. The only possible explanation thereof would be the influence of the cylinder walls, but this influence, in general, is a more complicated phenomenon than is shown. Nothing is mentioned in regard to the type of engine, and it is therefore not possible to analyze the probable performance of it. It is thought, however, that in a high-efficiency engine of this kind, the influence of the cylinder walls would not be great enough to be included in a diagram used for comparative purposes. The temperature-entropy diagram would be more appropriate for showing a cyclic variation of this kind.

Referring to the comparison in Par. 48 between turbine- and engine-driven generators, a rather inefficient turbine has been used against the very best kind of engine. A turbine can very easily be built for the given steam conditions with a water rate at its best load point of less than 50 lb. The desire expressed that turbine designers in this country should produce high-back-pressure units, similar to those built abroad, can be met at any time. The reason that European designers have led in this respect is simply that their fuel costs are much higher than in this country, and they can therefore justify a higher initial cost, which is inevitable with increased efficiencies.

The question of how to obtain pure feedwater must be settled for each individual case, and no general statements can be made. When the make-up water reaches and exceeds fifty per cent, as is often the case, it is doubtful if evaporators will be the most economical solution.

A. J. GERMAN.[1] The particular industrial plant that the author had in mind when preparing his paper is one in which the steam pressure is quite high for process or industrial purposes, and, therefore, one in which a high initial pressure is necessary in order to extract the proper amount of energy with a non-condensing turbine. As has always been the case, each particular plant must be worked out separately.

The writer wishes to discuss an industrial plant located in New England, where an initial pressure of 400 lb. per sq. in. would seem to be sufficiently high to bring about the economies looked for by the author. At present, in the plant in mind, there are two power stations. The more recent one carries a steam pressure of 250 lb. per sq. in. with a superheat of slightly over 100 deg. fahr. The present load demand amounts to 9000 kw., which is generated with condensing turbines. The older plant carries a pressure of 150 lb. per sq. in. and a superheat of 100 deg. fahr., and utilizes a stand-by turbine and an engine, with a large number of air compressors and other steam-driven auxiliaries. In the summer time nearly all of this machinery can be operated condensing if necessary, but for about seven months it is operated non-condensing, and the exhaust steam is used for the heating system, the plant operating at about 1 lb. back pressure.

In the 250-lb. plant a 750-kw. turbine previously run condensing was changed over to a back-pressure turbine for the heating system of that section of the factory. This turbine was a two-stage machine, but the second-stage wheel was removed, thereby obtaining a better water rate. Previous to using the exhaust steam for these turbines and auxiliary machinery for the heating system, hot-water heating boilers in both plants were necessary. The economies obtained are gratifying and found to be extremely flexible.

There is now under consideration the possibility of replacing the 150-lb. pressure boilers with boilers to operate at 400 lb. pressure and a total steam temperature of 750 deg. fahr.

The present pressure requirements for industrial steam vary. The object is to make an attempt to even up these pressures before turbine specifications can be worked out. Two different plans of turbine set-up are under consideration. One plan calls for the installation of one turbine, bleeding at 110 lb. pressure and exhausting at from 5 to 10 lb. The 110-lb. steam can be used in all the steam-driven compressors and other auxiliary machinery, thereby extracting energy from the steam, first in the turbine, and then in the compressors; the exhaust steam from these compressors and auxiliaries can be used up in the open feedwater heaters, in closed heaters for the hot-water supply to the factory, of which

[1] Chief Engineer, Scovill Manufacturing Co., Waterbury, Conn. Mem. A.S.M.E.

a great quantity is being used at all times, and also in the present hot-water heating system in the winter time. In the summer time the auxiliary machinery would furnish more exhaust steam than would be needed, and it therefore would have to operate condensing. The 5 to 10 lb. back pressure from the turbine would be used for industrial steam and to heat the hot-water supply.

It cannot be expected that the industrial plant could have a set-up of power requirements and steam requirements that would remain uniform in proportion for the various seasons of the year. It is therefore necessary that such plants should generate that amount of power with back-pressure turbines as will furnish sufficient steam for industrial purposes and heat, the remainder of the power demand being supplied by condensing turbines.

The author notes that plants requiring considerable power rarely have an unlimited supply of cold water for condensing purposes. This is another factor of great importance in the use of higher steam pressures with non-condensing turbines. The load on the condensing turbines is decreased to the extent of the electric power generated by the non-condensing turbines, thus permitting a better vacuum and a lower water rate on the condensing turbines, especially in the summer season when the supply of cold water for condensing purposes is at its lowest.

When industrial steam and steam for heat is being produced at a lower cost by having it first pass through back-pressure turbines, the question arises as to whether the power plant or the department using the steam will be credited with this lower cost. In the plant just discussed each department is charged monthly with a net cost of power, hot water, air, heat, etc. Would it not be better to work out a plan whereby both power and steam for industrial purposes would benefit by this reduced cost on a pro-rata basis?

W. G. DIMAN.[1] In regard to the value of higher steam pressures in industrial plants the writer agrees that the thermal efficiency will be increased, but before any plant is laid out a close study should be made of the use of process steam, coördination of the load, and the heat use in connection with the manufactured product. When steam is used for power, the by-product heat for process work, and the heat from spent liquors used in heating new liquors, there is not much more that can be done, and it will be found that no cheaper power is available. When this condition exists nothing will be gained by using any higher pressure than is necessary to accomplish the results.

The fact must never be lost sight of that the cost of a kilowatt includes interest, taxes, money set aside for renewals, and all

[1] Superintendent of Power, Amoskeag Mfg. Co., Manchester, N. H. Mem. A.S.M.E.

money used for any purpose in the production of power. These are just as tangible elements of the cost of a kilowatt as the money that must be paid for fuel to generate that kilowatt. When these facts are considered the average public-service plant cannot produce a kilowatt for much less than two cents. While the current may be sold for less than this to an industrial plant, the loss must be made up through a higher rate schedule for other purposes.

When it is merely a question of first cost versus fuel economy, the intelligent choice must always be made by balancing the additional overhead costs entailed by the more efficient design against the obtainable savings. In this case some designers seem to have paid more than was warranted for extremely low B.t.u. consumption. The main problem in designing a plant is the best return on the investment coupled with reliability.

When comparing central-station and industrial-plant pressures and temperatures it will be found that central-station pressures average about 350 lb. and industrial-plant pressures about 200 lb.

Steam temperatures in the central plants will run about 200 deg. fahr. higher than temperatures in industrial plants. The central station, as a rule, will have the latest refinements for conserving heat, while in the industrial plant some are omitted. This might in some cases be a mistake, but in many cases where attempts have been made to follow central-station practice the expected results have not been realized. Central stations are much larger in capacity than industrial plants and small percentage savings mean a marked reduction in operating costs. Being larger in size and requiring a smaller number of operators, a higher grade of men is generally used and closer supervision is given. The equipment, as a rule, receives more systematic and skilled attention than is generally the case in the average industrial plant, resulting in a considerable reduction in operating costs.

Each plant has entirely different service requirements. Central stations are designed for the single purpose of producing electric power. Even under the best conditions it is not an efficient operation, as a large percentage of the heat is wasted before it is delivered at the point where it can be utilized. Increasing the pressures and temperatures reduces the losses, but even under the most extreme conditions of pressures and temperatures the losses are reduced only by a small percentage.

Industrial plants, on the other hand, in most cases supply both heat and power. During the heating season, the demand on the power plant for heat frequently is greater than the demand for power. This condition presents an entirely different problem and is one of the main reasons for the wide difference in operating pressures and temperatures. With a heating load the temperatures required at the point where the heat is used generally are comparatively low, so that there is nothing gained by increasing pressures and temperatures beyond those required for heating or proc-

ess work as far as efficiency is concerned. Just as much heat can be transferred at low pressures and temperatures as at high temperatures and pressures, and a boiler plant can operate as efficiently at low as at high pressure.

If the demand for low-pressure steam for heating and process work approximately equals the output of the boiler plant, it is necessary only to increase the pressures sufficiently to produce the required power. It is only where the yearly steam demand for power greatly exceeds the demand for heating and process work that higher pressures and temperatures become worthy of consideration. If the amount of exhaust steam from the power-generating units is practically all utilized, increasing the efficiency of these units would reduce the output of the exhaust, and additional steam would have to be supplied at reduced pressures direct from the boilers.

There is installed and in operation at the plant of the Amoskeag Manufacturing Company a 5000-kw. straight non-condensing unit from which is drawn all of the steam used in the worsted department. Last year this unit developed about 5,900,000 kw-hr., when operating with steam at 150 lb. per sq. in. at the throttle against a 27-lb. back pressure. The unit is designed to operate with 225 lb. at the throttle against the same back pressure. While considerably more load can be carried by the same amount of steam passing through the unit, the interest on the investment necessary to furnish the higher steam pressure would offset any saving gained by the additional load carried.

High pressures and temperatures mean special equipment, high-grade construction, increased investment costs, higher fixed charges, and generally increased trouble and higher maintenance charges. In the average industrial plant moderate pressures and temperatures will generally be found to be the most efficient. Only in exceptional cases will the extreme high pressures and superheat prove of any material advantage.

Almost every industrial plant offers opportunity for utilizing waste heat or for producing power as a by-product of the process. There is a large field, much of which is untouched, in which central-station experience gives but little help. Successful applications are invariably the result of long experience, intimate and thorough knowledge of the manufacturing process, and careful study. One of the principal opportunities today is a further development in the use of heat in manufacturing processes. More efficient industrial apparatus can be furnished, and instead of furnishing a machine simply to dry materials, manufacturers can furnish one to dry a certain amount of material with a guaranteed steam consumption. Industrial executives would do well to give considerable thought to this whole subject. If advantage is taken of the possibilities a large saving will be made.

F. A. WEITSTEIN.[1] The author mentions repeatedly that high-pressure boilers in connection with non-condensing turbines may be used to great advantage in industrial plants, provided such plants are working at a high load factor. The load factor of an industrial steam and power plant is indeed of much greater importance in the application of high-pressure boilers and non-condensing turbines than has generally been recognized. A low load factor, that is, a fluctuating demand for process steam, will render the operation of high-pressure boilers very difficult. This is due to the fact that the storage capacity of modern high-pressure boilers compared with the old return tubular boilers, with their large water space, is practically negligible, so that these boilers can only with great difficulty follow sudden fluctuations in the demand for steam. Fluctuating demand for process steam, further, very seriously affects the operation and efficiency of non-condensing turbines. It is characteristic of non-condensing turbines that their efficiency decreases much more rapidly with decreasing steam flow than is the case in straight condensing turbines. In many industrial plants, therefore, the average efficiency obtained in non-condensing turbines is very low. Frequently the advisability of installing non-condensing turbines may be questioned, due to the wide difference that exists in the fluctuating demand for process steam and the demand for power.

The development of the steam accumulator system, which not only makes it possible to overcome the above-mentioned difficulties but at the same time is of additional advantage for the manufacturing plant, has therefore come at an opportune time. The accumulator makes it possible to operate the boiler plant at a load factor of practically 100 per cent and allows a constant flow to pass through the non-condensing turbines. These latter, therefore, will work with maximum efficiency. While without an accumulator steam often has to be by-passed to the turbines and at other times too little process steam is required for generating the required amount of power, the accumulator makes it possible to pass all of the process steam through the turbines and to generate a maximum amount of cheap non-condensing power. The accumulator, in fact, renders the boiler plant, the power plant, and the manufacturing plant to a great extent independent of each other, making possible the most intensified and efficient operation of all of them. It will therefore greatly facilitate the application of high-pressure boilers and non-condensing turbines to all industrial plants with fluctuating steam demand.

H. G. BARNHURST.[2] The paper is devoted mainly to boiler rather than to furnace operation. The only effect that higher

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boiler pressures would have on a pulverized-coal-fired furnace would be the influence of higher pressures on the design of water-tube furnace construction. Pulverized-coal-fired furnaces are ordinarily designed to operate at high rates of evaporation. The water tubes in the furnace walls, being exposed to direct radiant heat, work at higher rates of evaporation than most of the boiler tubes. Deposits of scale to a degree unobjectionable in boiler tubes may therefore introduce danger of tube failure when occurring in furnace-wall tubes. The tube failure in a furnace is obviously of increasing seriousness as the pressures increase. These factors warrant careful consideration of the employment of water tubes under high boiler pressures in furnaces intended to be operated at high rates of evaporation.

Air-cooled walls are being used, of designs that give entirely satisfactory service under these conditions of high evaporation and with consequent higher furnace temperatures. Experience with water tubes in furnaces has been limited to what are today relatively moderate pressures, although evaporative rates of the highest have been obtained.

Experience under these conditions does not indicate that there is apt to be serious trouble from such tube failure, although this condition should be borne in mind in designing furnaces for use with boilers of pressures above 400 lb. per sq. in. This caution should apply particularly to boilers in industrial plants where the make-up percentage is often higher than in central stations.

It might be interesting to mention that the boilers in the new Columbia Power Station, at Miami Fort, Ohio, are built for 600 lb. pressure. The boiler installation consists of eight units, six standard and two reheat boilers. These boilers are fired with pulverized coal and are equipped with air-cooled walls. The furnaces are designed for continuous operation at between 300 and 400 per cent of rating.

THE AUTHOR. The discussion of turbine efficiencies has been illuminating. Mr. Dahlstrand's conclusion that a high back-pressure turbine need be no less efficient than a condensing machine seems to be supported by the other designers who have discussed this paper. While it is true that the coefficient of friction increases with the increase in density and viscosity of steam, with high pressures the blade and nozzle areas swept by the steam are reduced. The net result may be a reduction of friction losses large enough to offset the relatively greater leakage. At any rate, American manufacturers are now offering high-back-pressure machines having a maximum guaranteed efficiency, relative to the Rankine cycle, of more than 75 per cent. This is materially higher than was assumed in this paper, and it means that a greater electrical output per unit quantity of steam may be obtained, or that for a given output a lower initial pressure may be used.

These high efficiencies have been quoted on machines ranging from 3000 to 5000 kw. capacity. The turbine efficiency in this high-pressure range is materially less for smaller-sized units, and for a very small installation better results can be obtained by using a reciprocating engine.

It has been pointed out several times in this discussion that the proposed methods of power generation can be applied economically in only a limited number of industrial plants. It is equally true, perhaps, that only a limited number of industrial plants should generate electric power at all. Of those which have any justification for generating power, a fairly high percentage can afford to do so with higher steam pressures than are now common.

Mr. Jones has raised an objection to piping very high-pressure steam for long distances. The objection is well founded. Undoubtedly it is better to locate high-back-pressure prime movers as close to the boiler as practicable and transport the steam at moderate pressures.

The comment on feedwater treatment is very reassuring, but this remains the most serious problem in connection with the adoption of high steam pressures in non-condensing plants.

Mr. Randall's comment on cycle efficiencies is not really at variance with the statements made in the paper. If Mr. Randall will attempt to attack the problem in the same way that it was solved for the condensing plant by Messrs. Hirshfeld and Ellenwood,[1] he will appreciate the force of the statement that the cycle efficiency cannot be made the basis of calculations for the non-condensing plant.

The author agrees with Professor Ellenwood that it should be possible to build a high-pressure steam engine in which the radiation loss would be less than that indicated by the figures in the paper. Manufacturers have not, perhaps, appreciated the importance of this item in a high-back-pressure unit.

The slope of the expansion lines for the steam engine, as given in Fig. 3, has troubled many. It represents, however, the best estimate of the designers of machines for this service, and there is nothing irrational about it. For a perfect prime mover the expansion line on the temperature-entropy chart would be vertical. Leakage and friction tend to force the line to the right, while radiation tends to force it to the left. In steam engines, as now designed for the higher back pressures, radiation is the predominant factor, amounting, as Professor Ellenwood points out, to 20 per cent of the energy delivered by the engine. This should not be confused with the energy delivered *to* the engine, which is the basis on which the percentage of radiation is usually figured for a con-

[1] "High Pressure, Reheating, and Regenerating for Steam Power Plants," by C. F. Hirshfeld and F. O. Ellenwood, Trans. A.S.M.E., vol. 45 (1923), p. 663.

densing machine. On the latter basis, the radiation would ordinarily be less than 3 per cent.

Fig. 9 has been questioned because it shows a higher water rate, and at the same time a higher thermal efficiency, for a turbine than for a reciprocating engine. The explanation of this seeming paradox is found in the quality of the exhaust steam, also shown in Fig. 9. While the turbine uses more pounds of steam per unit of output, the energy extracted from each pound is so much less that the total energy consumption per unit of output is also less. The excessive radiation of the engine unit is responsible for this condition. This is another illustration of the fundamental difference in considering "efficiencies" in the industrial plant and the central station. Provided that the water rate was less, high radiation losses would be no disadvantage for a condensing unit.

Some concern has been expressed over the lack of skilled operators in the industrial power plant. This point of view can be readily understood, but one should not compare the very largest and best central stations with the smallest and meanest of industrial plants. Plant for plant, the industrial personnel will not be found so far inferior as is commonly supposed. Judged by the three marks of good operation — safety, continuity of service, and low cost of output — the leading industrial plants will hold their own with the leading central stations. One plant in particular has a record of only one interruption to service in 40 years' operation. A ruptured ear drum is the most serious accident that has ever occurred to any employee at that plant. There are few central stations which can approach that record for safety or service; there are likewise very few which equal this plant in the low cost of steam production.

In particular reference to high pressures and temperatures, there are many industrial plants which are using higher pressures and higher temperatures in their processes than have been seriously considered by any public utility. In the majority of industrial plants which have the load conditions that make high steam pressures economical, it is probable that the operating personnel will be found entirely adequate.

In conclusion, the author wishes to second Mr. Dickinson's appeal for better operating data. Complete and accurate tests of commercially operating power-plant equipment are all too few. The early publication of test data relating to the operation of high-pressure equipment now in service would be extremely valuable to engineers who are contemplating installations along the lines proposed in this paper.

No. 1982

HEAT-TREATMENT DATA ON QUALITY STEEL CASTINGS

By Albert E. White,[1] Ann Arbor, Mich.

Member of the Society

This paper summarizes a study which has been made on heat-treatment practice for quality steel castings. It gives the results of laboratory tests on dendritic and dendrite-free steels after an annealing, a normalizing and drawing, and a spheroidizing treatment. It likewise records a large number of plant tests. The results show that a normalizing and drawing treatment gives superior results to those obtainable by annealing or spheroidizing.

STEEL castings are important in many major engineering operations. As connecting links between the boilers and the turbines in the large generating stations of the present day they perform one of their most outstanding functions. Castings for this service must be sound and possess high strength combined with good ductility. The three essentials necessary for the procurement of these properties are:

(*a*) Suitable composition
(*b*) Correct conditions surrounding the casting of the metals
(*c*) Proper heat treatment.

The last is the one with which this paper will mainly deal.

Chemical Composition

2 For most steel-casting purposes straight carbon-steel castings are adequate. Alloy-steel castings may be used if properties superior to those obtainable in straight carbon-steel castings are desired. The analysis limits for straight carbon-steel castings should be as follows:

Carbon, any 10-point range desired between 0.15 and 0.50 per cent
Manganese, 0.50 to 1.00 per cent
Silicon, above 0.20 per cent
Sulphur, not above 0.05 per cent
Phosphorus, not above 0.05 per cent.

[1] Professor of Metallurgical Engineering, Director, Department of Engineering Research, University of Michigan.

Contributed by the Power Division and presented at the Annual Meeting, New York, November 30 to December 4, 1925, of The American Society of Mechanical Engineers.

3 Objection may be raised against a ten-point carbon range for steel castings. Properly operated plants should find no difficulty in keeping within ten points. Scientific heat treatment incidental to the general manufacturing operation requires a limitation of the carbon to a ten-point range if best results are to be obtained. The limitations for manganese are not so vital. Enough must be present properly to deoxidize the steel and to convert all of the ferrous sulphide into manganous sulphide. If the usual methods of heat treatment were employed, too much manganese would result in the production of a steel which would have too little ductility.

4 Steel castings should always have their silicon content above 0.20 per cent. If less, the steel would, without doubt, be wild when poured. If so, the castings would be full of blow holes and other imperfections. Sulphur and phosphorus should both be low. These elements are detrimental in the extreme in steel castings and every effort should be made to keep them at a minimum.

Conditions Surrounding Casting

5 Chapters might be written with regard to the correct conditions surrounding the casting of the metal. Some of the most outstanding of these factors are:

(*a*) The temperature of the metal during the pouring
(*b*) The rate of cooling
  a From the liquid to the solid state
  b From the solidification temperature to atmospheric temperature
(*c*) The methods of gating
(*d*) The size, number, and distribution of the risers
(*e*) The size of the casting
(*f*) The design of the casting
(*g*) The kind of mold.

Heat Treatment

6 This paper is mainly written, however, to discuss the influence of heat treatment on the physical characteristics of steel castings. For that reason the above items will not be developed. They are, though, of paramount importance. In fact, quality steel castings cannot be produced without due and proper regard for each and every one of the above items.

7 Heat treatment may consist of

(*a*) The ordinary furnace anneal
(*b*) An air quench and a draw
(*c*) A spheroidizing anneal
(*d*) An oil or water quench and a draw.

8 Other treatments or modifications of the above treatment might be employed; if so, they would be the exception and not

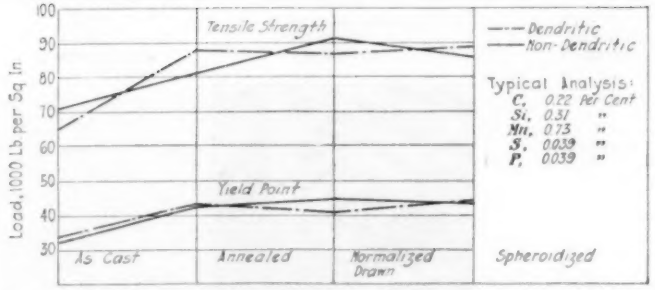


FIG. 1 RELATION BETWEEN THE TENSILE STRENGTH COUPLED WITH THE YIELD POINT AND THE HEAT TREATMENT OF CAST STEEL

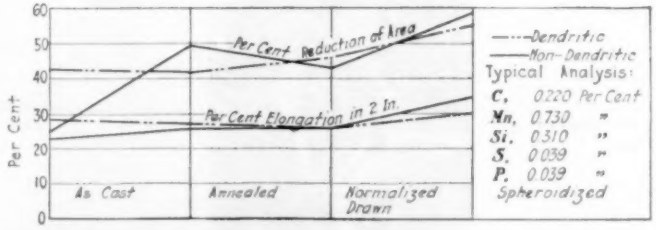


FIG. 2 RELATION BETWEEN THE PERCENTAGE OF ELONGATION COUPLED WITH THE PERCENTAGE OF REDUCTION OF AREA AND THE HEAT TREATMENT OF CAST STEEL

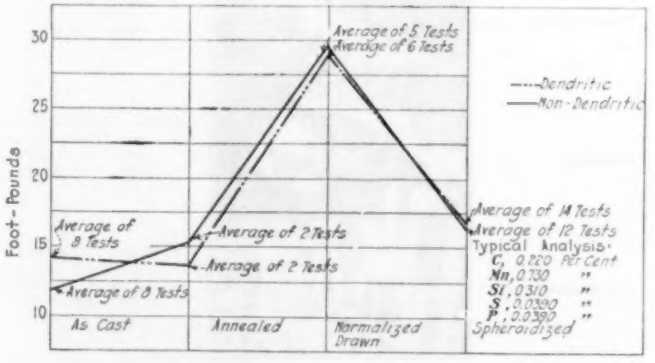


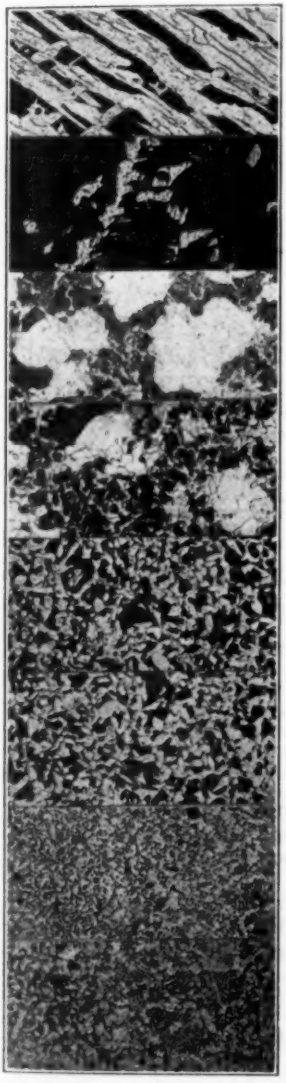
FIG. 3 RELATION BETWEEN IMPACT STRENGTH AND HEAT TREATMENT OF CAST STEEL

TABLE 1 SHOWING THE RELATION BETWEEN TYPE OF STRUCTURE, HEAT

Bar No.	Initial condition	Microstructure before treatment	Heat treatment
13	Dendritic.		(1) No treatment.
14	Non-dendritic.		(1) No treatment.
1	Dendritic.		(2) Heated at 1760 deg. fahr. for 7 hr. Furnace-cooled.
2	Non-dendritic.		(2) Heated at 1760 deg. fahr. for 7 hr. Furnace-cooled.
5	Dendritic.		(3) Heated at 1760 deg. fahr. for 7 hr. Air-cooled. Heated at 1100 deg. fahr. for 6 hr. Air-cooled.
6	Non-dendritic.		(3) Heated at 1760 deg. fahr. for 7 hr. Air-cooled. Heated at 1100 deg. fahr. for 6 hr. Air-cooled.
9	Dendritic.		(4) Heated at 1760 deg. fahr. for 7 hr. Air-cooled. Heated at 1275 deg. fahr. for 30 hr. Air-cooled.
10	Non-dendritic.		(4) Heated at 1760 deg. fahr. for 7 hr. Air-cooled. Heated at 1275 deg. fahr. for 30 hr. Air-cooled.

(1) As cast. (2) Annealed. (3) Normalized and drawn. (4) Spheroidized.

TREATMENT, AND PHYSICAL PROPERTIES FOR MEDIUM-CARBON CAST STEEL

Microstructure after treatment	Tensile strength, lb. per sq. in.	Yield point, lb. per sq. in.	Elongation in 2 in., per cent	Reduction of area, per cent	Impact strength, ft-lb.
	65,500	34,100	28.13	41.80	14.25 Bar No. 15
	71,300	32,600	23.28	25.61	11.80 Bar No. 16
	88,000	43,700	27.00	41.60	13.50 Bar No. 3
	81,200	42,800	26.00	49.00	15.25 Bar No. 4
	86,650	41,000	26.00	46.26	29.00 Bar No. 7
	91,350	44,700	26.00	43.55	30.50 Bar No. 8
	78,600	44,600	30.50	56.50	16.55 Bar No. 11
	76,000	43,800	34.00	58.57	15.20 Bar No. 12

the rule. As a matter of fact, the ordinary furnace anneal is the one usually employed, though, as will be shown later, it is not the treatment which gives the best combination of strength and ductility for power-plant castings. Of the four treatments mentioned, the oil or water quench followed by a draw has not been

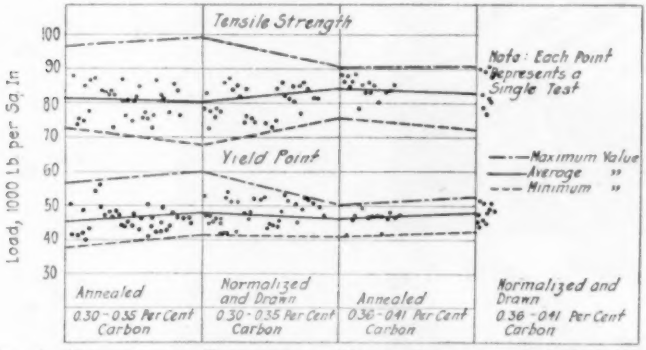


FIG. 4 RESULTS OF COMMERCIAL TESTS SHOWING THE RELATION BETWEEN THE TENSILE STRENGTH COUPLED WITH THE YIELD POINT AND THE HEAT TREATMENT OF CAST STEEL

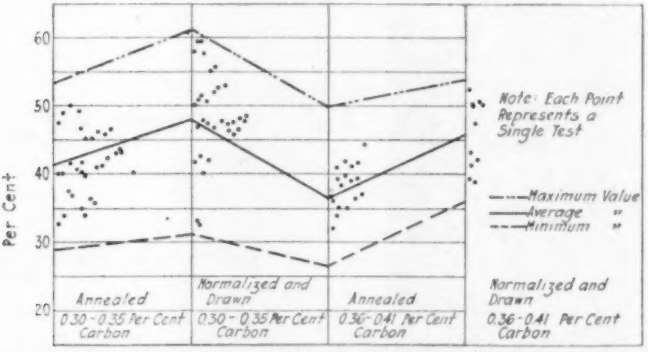


FIG. 5 RESULTS OF COMMERCIAL TESTS SHOWING THE RELATION BETWEEN THE PERCENTAGE OF REDUCTION OF AREA AND THE HEAT TREATMENT OF CAST STEEL

considered. This treatment is not applicable for castings of large size. Neither is it applicable for castings of intricate shape.

9 Careful studies were made of

(*a*) The ordinary furnace anneal

(*b*) An air quench followed by a draw

(*c*) A spheroidizing anneal.

These studies were made on both dendritic and non-dendritic steels for the purpose of ascertaining the effects of these treatments on these two types of steels.

10 In Table 1 are recorded the results of laboratory tests on dendritic and non-dendritic manifesting steels in the untreated, annealed, normalized and drawn, and spheroidized condition. Figs. 1 and 2 were drawn from the values given in Table 1. They show the effects of the different heat treatments on the tensile strength, yield point, reduction of area, and percentage of elongation. Fig. 3

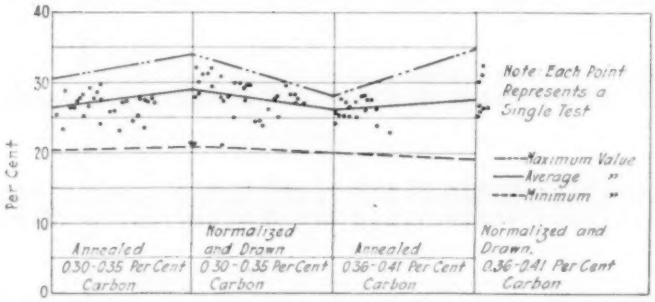


FIG. 6 RESULTS OF COMMERCIAL TESTS SHOWING THE RELATION BETWEEN THE PERCENTAGE OF ELONGATION AND THE HEAT TREATMENT OF CAST STEEL

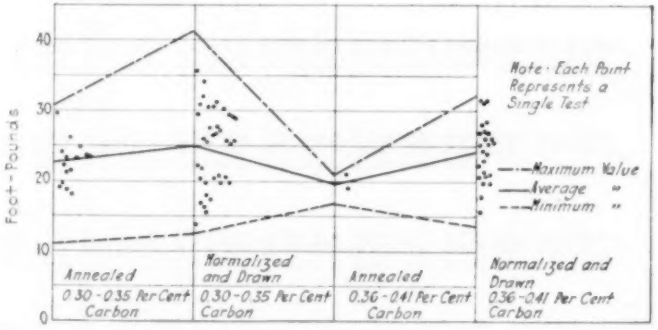


FIG. 7 RESULTS OF COMMERCIAL TESTS SHOWING THE RELATION BETWEEN THE IMPACT STRENGTH AND THE HEAT TREATMENT OF CAST STEEL

shows the effect of different laboratory heat treatments on the impact value of steel castings.

RESULTS OF LABORATORY TESTS

11 These tests demonstrated the following:

(*a*) The presence of dendrites makes but little difference from a tensile and impact test standpoint. This statement, however, does not imply an endorsement of dendrite-manifesting steels, for such steels are, for a number of reasons, undesirable.

TABLE 2 RESULTS OF COMMERCIAL TESTS ON STEEL CASTINGS ANNEALED BY CRANE CO. — CARBON CONTENT, 0.30 TO 0.35 PER CENT

Test No.	Mark	Chemical composition					Tensile strength, lb. per sq. in.	Yield point, lb. per sq. in.	Per cent elong. in 2 in.	Per cent red. of area	Izod impact value, ft-lb.
		Si.	Mn.	S.	P.	C.					
1	4	0.38	0.76	0.020	0.028	0.33	88,200	50,100	25.5	39.8	....
2	3474	0.36	0.77	0.020	0.029	0.30	75,000	41,700	26.5	40.4	....
3	3499	0.33	0.77	0.018	0.027	0.34	76,000	41,650	28.5	44.0	....
4	3501	0.33	0.76	0.021	0.023	0.31	75,800	41,800	26.5	36.6	....
5	3504	0.30	0.75	0.017	0.026	0.33	85,300	48,800	23.5	41.3	....
6	3506	0.28	0.72	0.020	0.037	0.31	78,900	42,800	27.0	46.8	....
7	1	0.32	0.75	0.020	0.026	0.30	86,800	54,650	27.5	49.9	29.4
8	24	0.39	0.77	0.024	0.043	0.35	96,500	55,100	20.5	28.9	....
9	5	0.38	0.76	0.021	0.020	0.35	87,300	49,600	26.5	37.5	22.0
10	6	0.34	0.68	0.018	0.038	0.35	83,550	47,000	27.5	41.6	24.0
11	11	0.30	0.66	0.019	0.019	0.31	82,950	48,050	28.0	40.4	....
12	20	0.30	0.70	0.025	0.050	0.35	82,700	46,800	25.0	39.5	....
13	21	0.35	0.72	0.024	0.044	0.34	83,650	48,300	26.5	42.8	....
14	26	0.35	0.71	0.018	0.042	0.31	82,850	47,750	29.0	45.5	....
15	2971	0.25	0.65	0.020	0.022	0.31	74,000	43,900	27.5	36.7	....
16	3	0.40	0.77	0.029	0.023	0.33	87,350	43,700	24.0	36.3	....
17	3330	0.31	0.74	0.018	0.025	0.35	80,500	44,200	27.5	41.0	....
18	3331	0.26	0.73	0.022	0.032	0.30	76,300	43,900	28.5	46.8	....
19	3269	0.41	0.74	0.018	0.038	0.31	80,900	47,800	27.5	38.3	....
20	3424	0.30	0.74	0.024	0.035	0.31	75,000	42,600	29.5	46.4	25.9
21	3427	0.33	0.76	0.021	0.035	0.33	80,200	46,500	26.0	41.6	23.0
22	3619	0.32	0.76	0.019	0.032	0.34	81,500	44,900	26.0	42.3	19.0
23	3244	0.28	0.72	0.026	0.042	0.31	85,000	50,100	23.0	34.0	....
24	3428	0.31	0.80	0.024	0.037	0.33	76,000	42,300	27.0	43.1	21.2
25	3429	0.30	0.80	0.020	0.023	0.34	77,800	43,150	27.5	43.4	24.7
26	3450	0.34	0.75	0.028	0.027	0.34	75,800	40,000	24.5	32.6	18.2
27	3453	0.34	0.72	0.023	0.021	0.30	73,200	37,750	30.5	43.4	19.2
28	3459	0.37	0.80	0.021	0.036	0.31	76,650	43,350	25.5	40.1	11.0
29	3756	0.35	0.74	0.019	0.024	0.33	86,900	49,650	23.5	33.8	18.7
30	3759	0.32	0.72	0.025	0.037	0.32	82,600	46,600	27.5	49.0	23.5
31	3762	0.28	0.75	0.022	0.034	0.30	82,200	42,100	27.5	46.0	22.5
32	3765	0.37	0.79	0.027	0.029	0.30	78,050	44,900	30.5	53.4	30.3
33	3212	0.35	0.79	0.025	0.040	0.32	84,800	48,150	27.0	35.0	....
34	3759	0.33	0.75	0.023	0.035	0.34	82,600	46,600	27.5	49.0	23.5
35	3881	0.32	0.73	0.021	0.037	0.35	77,100	39,500	27.0	38.7	21.3
							81,141	45,307	26.37	41.3	22.2

TABLE 3 RESULTS OF COMMERCIAL TESTS ON STEEL CASTINGS ANNEALED BY CRANE CO. — CARBON CONTENT, BETWEEN 0.36 TO 0.41 PER CENT

Test No.	Mark	Chemical composition					Physical properties				
		Si.	Mn.	S.	P.	C.	Tensile strength lb. per sq. in.	Yield point, lb. per sq. in.	Per cent elong. in 2 in.	Per cent red. of area	Izod impact value, ft-lb.
1	7	0.31	0.71	0.015	0.029	0.38	89,200	50,600	24.0	31.8	....
2	3482	0.33	0.72	0.024	0.027	0.37	76,800	40,100	27.5	40.8	....
3	3496	0.31	0.74	0.020	0.030	0.36	78,800	42,100	27.0	41.6	....
4	19	0.30	0.73	0.017	0.013	0.36	84,700	45,900	25.5	38.5	....
5	8	0.32	0.68	0.016	0.024	0.39	87,800	47,600	25.5	39.7	21.0
6	10	0.32	0.73	0.015	0.021	0.39	89,300	50,250	26.0	39.4	21.0
7	17	0.33	0.74	0.017	0.048	0.40	90,700	49,750	25.5	36.3	....
8	3326	0.35	0.72	0.018	0.054	0.40	85,500	46,500	27.0	39.7	....
9	3198	0.36	0.77	0.029	0.038	0.38	82,750	47,050	28.0	41.3	....
10	3254	0.43	0.71	0.020	0.029	0.38	86,000	47,300	26.0	39.4	....
11	3198	0.36	0.77	0.029	0.038	0.38	82,750	47,050	28.0	41.3	....
12	3275	0.39	0.75	0.049	0.029	0.39	85,400	43,050	24.0	33.8	....
13	3433	0.30	0.77	0.020	0.026	0.37	80,050	46,100	27.5	49.9	19.0
14	3209	0.34	0.75	0.022	0.035	0.37	83,500	47,600	20.0	26.2	....
15	3241	0.38	0.76	0.017	0.028	0.36	84,000	46,000	27.5	44.6	....
16	3254	0.43	0.71	0.020	0.029	0.38	86,000	47,300	26.0	39.4	....
17	3874	0.27	0.78	0.022	0.022	0.36	88,200	44,400	22.5	36.4	17.0
							84,809	46,391	24.06	36.07	19.5

TABLE 4 PART 1 RESULTS OF COMMERCIAL TESTS FOR STEEL CASTINGS NORMALIZED AND DRAWN BY CRANE CO. — CARBON CONTENT, 0.30 TO 0.35 PER CENT

Test No.	Mark	Chemical composition					Tensile strength lb. per sq. in.	Yield point, lb. per sq. in.	Per cent elong. in 2 in.	Per cent red. of area	Izod impact value, ft-lb.
		Si.	Mn.	S.	P.	C.					
1	C1	0.32	0.75	0.020	0.026	0.30	78,700	52,200	30.0	58.4	29.5
2	D4	0.38	0.76	0.020	0.028	0.33	82,650	47,450	28.5	46.9	22.0
3	D11	0.36	0.77	0.020	0.029	0.30	77,800	44,800	31.0	50.3	31.2
4	D15	0.33	0.76	0.021	0.023	0.31	76,300	45,600	31.0	50.8	30.9
5	D14	0.33	0.77	0.018	0.027	0.34	78,550	46,200	29.5	48.9	34.3
6	D16	0.30	0.75	0.017	0.026	0.33	77,900	45,200	28.5	51.4	21.6
7	D3506	0.28	0.72	0.020	0.037	0.31	68,500	41,500	32.0	55.4	14.0
8	B5	0.38	0.76	0.021	0.020	0.35	85,900	51,600	27.5	48.2	23.0
9	B6	0.34	0.68	0.018	0.038	0.35	86,950	53,050	27.0	42.5	26.6
10	B11	0.30	0.66	0.019	0.019	0.31	80,200	50,100	31.0	50.9	30.8
11	B20	0.30	0.70	0.025	0.050	0.35	84,000	41,800	28.0	42.5	16.0
12	B21	0.35	0.72	0.024	0.044	0.34	84,300	41,600	25.0	46.5	15.0
13	B26	0.35	0.71	0.018	0.042	0.31	81,950	48,150	28.5	47.8	24.8
14	E2	0.25	0.65	0.020	0.022	0.31	75,800	46,000	30.0	47.8	30.4
15	E3	0.40	0.77	0.029	0.023	0.33	77,400	48,100	28.0	45.7	18.0
16	E3330	0.31	0.74	0.018	0.025	0.35	84,200	51,400	33.5	59.2	35.5
17	E3331	0.26	0.73	0.022	0.032	0.30	71,350	45,200	27.5	48.4	30.4
18	F3424	0.30	0.74	0.024	0.035	0.31	76,600	51,450	30.0	52.4	41.2
19	F3427	0.33	0.76	0.021	0.035	0.33	75,850	51,700	29.0	53.1	24.5
20	F3619	0.32	0.76	0.019	0.032	0.34	82,650	44,000	21.0	32.5	20.3
21	G3428	0.31	0.80	0.024	0.037	0.33	74,500	45,100	29.5	55.3	27.8
22	G3429	0.30	0.80	0.020	0.023	0.34	72,000	44,800	33.5	59.4	20.2
23	G3450	0.34	0.75	0.028	0.027	0.34	72,800	44,800	29.5	49.0	20.3
24	G3453	0.34	0.72	0.023	0.021	0.30	74,800	45,300	33.5	51.4	25.2
25	G3459	0.37	0.80	0.021	0.036	0.31	84,500	48,000	27.5	44.2	27.7
26	3756H	0.35	0.74	0.019	0.024	0.33	86,600	53,300	26.0	44.4	23.2
27	3759H	0.32	0.72	0.025	0.037	0.32	82,500	45,900	24.5	41.7	19.9
28	$3759H_1$	0.32	0.72	0.025	0.037	0.32	81,350	48,750	28.0	49.5	19.8
29	$3759H_2$	0.32	0.72	0.025	0.037	0.32	85,600	50,100	25.0	41.4	22.3
30	3762H	0.28	0.75	0.025	0.034	0.30	80,800	45,100	24.5	42.0	22.6*
31	3765H	0.37	0.79	0.027	0.029	0.30	77,600	48,300	29.5	46.5	29.2
32	3212H	0.35	0.77	0.030	0.040	0.33	85,600	51,900	24.0	32.9	27.0
33	3759I	0.33	0.75	0.023	0.035	0.34	86,500	51,000	27.0	43.4	24.5
34	3759I	0.33	0.75	0.023	0.035	0.34	83,900	50,300	27.0	45.5	24.7
35	3881I	0.32	0.73	0.021	0.037	0.35	81,550	49,450	28.5	46.9	31.0
36	3885I	0.29	0.71	0.021	0.032	0.35	81,400	47,100	28.5	47.4	26.2
							80,016	48,100	28.88	47.72	24.94

*Inclusion.

TABLE 4 PART 2 RESULTS OF COMMERCIAL TESTS FOR STEEL CASTINGS NORMALIZED AND DRAWN BY CRANE CO. — CARBON CONTENT, 0.30 TO 0.35 PER CENT

Test No.	Mark	Chemical composition					Tensile strength lb. per sq. in.	Yield point, lb. per sq. in.	Per cent elong. in 2 in.	Per cent red. of area	Izod impact value, ft-lb.
		Si.	Mn.	S.	P.	C.					
37	C24	0.39	0.77	0.024	0.043	0.35	99,550	60,400	21.0	30.9	22.9
38	D15f	0.33	0.76	0.021	0.027	0.31	.....	.....	....	....	28.8
39	Bf6	0.34	0.68	0.018	0.038	0.35	.....	.....	....	....	30.6
40	Bf5	0.38	0.76	0.021	0.020	0.35	.....	.....	....	....	23.2
41	Bf11	0.30	0.66	0.019	0.019	0.31	.....	.....	....	....	23.4
42	Bf20	0.30	0.70	0.025	0.050	0.35	.....	.....	....	....	28.8
43	Bf21	0.35	0.72	0.024	0.044	0.34	.....	.....	....	....	26.7
44	Bf26	0.35	0.71	0.018	0.042	0.31	.....	.....	....	....	29.5
45	E5f	0.41	0.74	0.018	0.038	0.31	.....	.....	....	....	25.0
46	F2	0.28	0.72	0.026	0.042	0.31	.....	.....	....	....	15.6
47	1H	0.30	0.81	0.026	0.028	0.33	.....	.....	....	....	18.3
						Miscellaneous tests					
1	F3026	0.30	0.70	0.026	0.040	0.21	69,350	46,250	34.0	61.7	27.8
2	F3425	0.30	0.73	0.020	0.032	0.29	76,650	51,750	32.0	53.4	32.0
3	C15	0.40	0.77	0.021	0.055	0.29	80,800	42,900	28.0	50.0	20.3
4	C1f	....	....	.....	.....	....	.....	.....	....	....	24.0
5	C2f	....	....	.....	.....	....	.....	.....	....	....	23.5
6	C3f	....	....	.....	.....	....	.....	.....	....	....	22.3
7	C4f	....	....	.....	.....	....	82,800	40,500	24.0	40.0	12.6
8	.....	....	....	.....	.....	....	.....	.....	....	....	....

TABLE 5 RESULTS OF COMMERCIAL TESTS FOR STEEL CASTINGS NORMALIZED AND DRAWN BY CRANE CO. — CARBON CONTENT, 0.36 TO 0.41 PER CENT

Test No.	Mark	Chemical composition					Tensile strength lb. per sq. in.	Yield point, lb. per sq. in.	Per cent elong. in 2 in.	Per cent red. of area	Izod impact value, ft-lb.
		Si.	Mn.	S.	P.	C.					
1	C10	0.32	0.73	0.015	0.021	0.39	83,350	47,150	29.6	51.7	23.0
2	D7	0.31	0.71	0.015	0.029	0.38	89,500	52,600	25.0	41.3	20.8
3	D12	0.33	0.72	0.024	0.027	0.37	82,800	44,000	27.0	47.6	25.7
4	D12f	0.33	0.72	0.024	0.027	0.37	.....	.....	....	....	24.5
5	D13	0.31	0.74	0.020	0.030	0.36	76,500	45,600	31.0	50.1	25.2
6	B19	0.30	0.73	0.017	0.013	0.36	75,700	42,200	32.0	50.4	18.2
7	Bf19	0.30	0.73	0.017	0.013	0.36	.....	.....	....	....	21.5
8	E17	0.33	0.74	0.017	0.048	0.40	88,950	50,600	26.0	42.8	20.9
9	E3826	0.35	0.72	0.018	0.054	0.40	72,600	48,950	34.5	50.4	31.4
10	G3433	0.30	0.77	0.020	0.026	0.37	87,500	51,100	25.5	39.1	28.7
11	G7	0.34	0.75	0.022	0.035	0.37	.....	.....	....	....	24.2
12	G8	0.34	0.75	0.022	0.035	0.37	.....	.....	....	....	14.0
13	G9	0.38	0.76	0.017	0.028	0.36	.....	.....	....	....	22.8
14	G10	0.38	0.76	0.017	0.028	0.36	.....	.....	....	....	32.2
15	F3	0.36	0.77	0.029	0.038	0.38	.....	.....	....	....	24.7
16	F6	0.39	0.75	0.040	0.029	0.39	.....	.....	....	....	15.6
17	3254H	0.43	0.71	0.020	0.029	0.41	90,400	52,800	19.0	36.0	28.6
18	$3254H_1$	0.43	0.71	0.020	0.029	0.41	.....	.....	....	....	31.3
19	3209H	0.34	0.75	0.022	0.035	0.37	.....	.....	....	....	26.7
20	3198H	0.36	0.77	0.029	0.038	0.38	.....	.....	....	....	25.3
21	2H	0.32	0.81	0.016	0.030	0.41	81,500	49,000	26.5	39.1	26.1
22	3874-I	0.27	0.78	0.022	0.022	0.36	80,156	49,100	30.0	53.7	25.7
23	3877-I	0.28	0.78	0.019	0.038	0.41	88,500	50,700	27.5	47.2	25.5
24	E4f	0.35	0.76	0.024	0.027	0.37	83,130	48,600	27.5	45.6	19.6
25	E6f	0.36	0.77	0.029	0.038	0.38	.....	.....	....	....	21.3
26	E7f	0.43	0.71	0.020	0.029	0.38	.....	.....	....	....	25.3
27	C8	0.32	0.68	0.016	0.024	0.39	.....	.....	....	....	31.3
											26.4

(*b*) The tensile strength, yield point, reduction of area, percentage of elongation, and impact properties are improved by any one of the three heat treatments.

(*c*) The spheroidizing treatment is superior to the annealing and normalizing treatments in so far as the reduction of area and percentage of elongation characteristics are concerned.

(*d*) The spheroidizing treatment is vastly inferior to the annealing and normalizing treatments when measured by the impact test.

(*e*) The normalizing treatment gives impact results approximately 100 per cent greater than any of the other treatments.

(*f*) Finally, by far the best all-around properties are obtained by the normalizing and drawing operation when the criterion is high strength, fair ductility, and maximum resistance to impact.

Plant Tests

12 These laboratory tests were augmented by plant tests on commercial runs of large power-plant castings. The results are recorded in Tables 2, 3, 4, and 5 and in Figs. 4, 5, 6, and 7. The tests were divided into two groups on the basis of carbon content; one group consisting of castings with from 0.30 to 0.35 per cent carbon and the other group with from 0.36 to 0.41 per cent carbon. These tests record values only after an annealing or a normalizing and drawing operation. They confirm beyond any question of a doubt, however, that normalized and drawn castings intended for power-plant purposes have properties superior to those produced by the standard anneal.

13 It is strongly recommended, therefore, that all large castings for power-plant purposes be heat treated by the normalizing and drawing method. Briefly this method consists in evenly heating the castings to between 1750 and 1800 deg. fahr., and holding them within this temperature range until uniformly heated. They should then be cooled to 100 deg. fahr., or below, in still air. The castings should then be uniformly heated to 1200 deg. fahr., after which treatment they may be cooled as desired.

14 With clean, sound castings of suitable composition thus treated, power plants and other users of quality steel castings would get the highest obtainable present-day value in steel castings.

Acknowledgment

15 The data in this paper were taken from a thesis on The Heat Treatment of Steel Castings, by Victor M. Mazurie. The work was done by Mr. Mazurie as holder of The Detroit Edison Fellowship in Metallurgy. Grateful acknowledgment is also made to the Crane Company through Mr. L. W. Spring. All of the data in Tables 2, 3, 4, and 5 were from tests made by the Crane Company on castings supplied to The Detroit Edison Company.

DISCUSSION

L. W. Spring.[1] This paper suggests a heat treatment, which, as far as steel castings for power-house work are concerned, is comparatively new. It certainly gives physical properties, especially as to yield point, reduction of area, and impact value, somewhat more desirable than those conferred by the older "dead anneal." The grain size and arrangement are particularly improved. One foresees little disadvantage, at least for fairly plain, sufficiently heavy castings. With stresses relieved by the "draw" at 1200 deg. fahr., they should retain their contour in service, except, perhaps, thin-walled, complicated, or delicately aligned castings, such as turbine castings, which, the writer understands, have always been "dead annealed" to avoid possible later distortion in service. Most of the castings referred to in the paper were heavy ells, tees, and reducers with wall thicknesses of two or three inches.

V. T. Malcolm.[2] Cast-steel valves and fittings designed for high-pressure lines, whether oil, gas, or steam, must conform to rigid specifications and are covered by strong guarantees. To stand back of these guarantees and to assume the risk implied, it is essential that the manufacturer be thoroughly familiar with every step in the manufacturing process, especially with respect to the heat treatment of castings, and that theory and practice be coördinated and methods developed to a high state of efficiency, in which quality is a paramount feature.

The author has touched on several points of especial interest, i.e., that to normalize and draw is the best treatment for steel castings; and, secondly, the detrimental effect of the dendritic structure. The writer agrees with him in these statements. In regard to the normalize and draw treatment, the writer's company carried out a series of investigations some four years ago, and from the result of these investigations we adopted this method of heat treatment and the results in service have been satisfactory.

Our investigations showed the dendritic structure to be one that should never exist in quality steel castings and this structure may be readily guarded against if proper precautions are taken. Metal at times is poured into sand molds at an incorrect or very high temperature and castings made with such metal are kept from cooling rapidly after solidification takes place, as the temperature is held within the granulating range for quite some time. This condition will cause the production of a coarse-grain structure known as "ingotism" and under which crystalline growth occurs by linear disposition. We then have the so-called dendritic struc-

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[2] Metallurgical Engineer, The Chapman Valve Mfg. Co., Indian Orchard, Mass.

ture, and any impurities contained in the steel are rejected to the grain boundaries and cannot be eliminated by heat treatment.

Of course the quality of the steel depends upon the selection of raw materials, the skill of the operators in melting, and the subsequent care in the heat-treating process. The great importance of clean, sound steel, properly heat treated, is now recognized by the fact that it is able to meet the exacting specifications of high-grade castings for valves and fittings in the modern power plant and oil refinery.

The production of clean, sound steel is the first requisite and the second and equally important requisite is the proper and thorough heat treatment of the castings, so that the steel may possess the best possible physical properties. High-grade steel castings may be poorer in quality through improper heat treatment than an inferior steel properly heat treated. Heat treatment, of course, will not give to steel properties which it does not contain, but it does bring out and make available properties that are inherent to the steel.

In a specialized class of work such as valves and fittings, the chemical analysis of the steel should be kept within certain limitations, so that the heat-treatment cycle once set may be maintained constantly. The reason for this is that steel confirming to certain chemical specifications is subject to a certain definite heat treatment with certain definite results as required by the physical specifications. Therefore, in order to obtain the best results and respond to heat treatment, steel should be poured at a correct temperature and be thoroughly deoxidized, as the presence of oxides, slag inclusions, etc., has a decided effect on the ferrite network and grain structure.

Without going into detail we may say that the purpose of heat treatment may be summed up as follows:

a To increase the softness and ductility

b To secure a desirable combination of strength and elastic limit with fair ductility and resistance to shock

c To eliminate the existing coarseness of grain and to impart to the steel a fine dense structure, insuring toughness

d To relieve any internal strains that may be caused by uneven contraction of castings in the mold.

The heat-treatment operations consist mainly of heating steel to the proper temperature, soaking at this temperature for the proper time and cooling from soaking temperature to the atmosphere. The first point to be considered is the temperature at which to heat the steel past the critical range so that the coarse structure may be broken down. The second point is the length of time that the castings should be kept at the required temperature. Complete saturation of castings is one of the most essential requisites toward good heat treatment, and prolonged soaking is preferable to incomplete saturation. The third point is the cooling

from soaking temperature, as the fine grain structure imparted to steel as noted in previous paragraphs, must be retained. Rapid cooling is, of course, the most effective means of retaining the fine grain structure, and when this is done, a drawing is necessary to produce the physical characteristics required. Slow cooling in the furnace from soaking temperature has been found to be unsatisfactory.

To sum up, it is necessary to have the correct soaking temperature, the correct cooling process, and at the same time to have absolute temperature control and uniformity of temperature throughout the charge of castings, as serious defects may result from lack of uniformity. The method of charging castings into the furnace and the manner of withdrawing them will have a great bearing on the uniformity of product. Finally, it may be stated that steel castings properly heat treated will possess physical characteristics comparable with heat-treated forged steel.

R. A. BULL.[1] References by the author to the chemical composition that should characterize "straight carbon-steel castings," prompt the writer of this discussion to submit a few comments. These are not offered by authority of the Joint Committee on the Investigation of the Effects of Phosphorus and Sulphur in Steel. They are made on his personal initiative by one who, as the representative of the American Foundrymen's Association on this Joint Committee, is informed regarding the results obtained to date from this coöperative research. It has been prosecuted by representatives of many national technical organizations, the War Department, the Navy Department, and the Bureau of Standards, whose director is chairman of the Joint Committee.

The work already done by and for the committee has covered several years, and involved many thousands of tests, duplicated for accurate checking purposes at Watertown Arsenal and Annapolis. Thus far the investigation has been confined to the effects of sulphur, the influence of which, on steel in service, has been conceded for years by most steel metallurgists to be obscure. This phase of the research has been the subject of five published preliminary reports, issued for the Joint Committee by the American Society for Testing Materials.

Work done by the Joint Committee thus far, while uncovering nothing as to the effects of phosphorus, throws considerable light on the influence of sulphur on mill steels of numerous grades and compositions. The phase of the research on steel castings is now being undertaken. In the temporary absence of reliable data regarding the product of the steel foundry, it seems unreasonable to assume that the effects of sulphur in finished steel castings under test or service conditions will be unlike those in rolled steel. It

[1] Director, Electric Steel Founders' Research Group, Chicago, Ill.

must be plain to every unbiased person who has analyzed the published data assembled by the Joint Committee, that there is yet no scientific basis for an unqualified statement that sulphur is decidedly injurious to steel castings and that every effort should be made to keep the content of this element at a minimum.

The Joint Committee has not thus far expressed its collective opinion of the results of its own work. It has followed the conservative policy of collecting and reporting data without forming official tentative conclusions. It can be safely predicted that when the committee's opinions are published, they will be found free from bias, and based only on evidence that is conclusive, accumulated for the purpose of making facts rather than prejudice or suspicion the foundation for engineering practice and its consistent demands on steel manufacture. It is hoped that engineers will refrain from forming conclusions which, like that of the author relating to sulphur, are not justified by anything yet ascertained by the Joint Committee as the result of its own work, or of the work of others at home or abroad thus far published, or otherwise made available to the committee, which has diligently sought information from every reliable source.

The author establishes 0.50 per cent and 1.00 per cent, respectively, as maximum limits for carbon and manganese, for what he terms straight carbon-steel castings, as distinguished from alloy-steel castings. He may feel that steel containing, for example, 0.55 per cent carbon or 1.05 per cent manganese, should not be classed as a straight carbon steel. Metallurgists naturally find difficulty in trying to agree as to where to make distinctions between some alloy or special steels, and carbon steel. While not wishing to take issue with the author as to the proper point at which to draw the line, the writer desires to point out, in order to prevent a possible misunderstanding, that considerable excellent steel for castings is purposely made to contain higher percentages of carbon or manganese than indicated in the paper; and that many do not regard steels containing these higher percentages as of the alloy class represented by steels containing nickel, chromium, etc.

Steel-foundry metallurgists are gratified to see valuable contributions such as the author has made, to add to the store of knowledge on the highly important subject of heat treatment. Fortunately this topic is one of increasing interest to engineers, who have not always realized its great significance.

THE AUTHOR. Mr. Bull's comments on the nature of the paper's reference to sulphur would lead one to believe that the upper limit on sulphur should either be removed or else materially raised. The author appreciates that there has been a consistent effort on the part of certain steel manufacturers, much if not all of which has been in good faith, to raise the rejection limit of sulphur. Some interesting data have been presented by the Joint Committee on

the Investigation of the Effect of Phosphorus and Sulphur in Steel. Though the information now available suggests the possibility of raising the rejection limit, many engineers are nevertheless loath to approve such a course, feeling that the data are not sufficiently comprehensive or sufficiently well seasoned to warrant such a step.

The author is not aware that such representative bodies as the American Society for Testing Materials or the Society of Automotive Engineers or other organizations interested in specifications have made any material, if any, modification of their sulphur limits in specifications. Also sulphur and phosphorus have been the two elements on which the two steel-castings committees of the American Society for Testing Materials have been in quite general agreement as to a desirability for limitation. No one, to his knowledge, believes sulphur adds desirable qualities to steel beyond those of free cutting, and free cutting is not a property upon which any emphasis is placed in steel castings.

The author grants that this may be libeling sulphur. It is not beyond possibility that sulphur may have been cavorting in bad company, but until there is further information on the subject than is at present available, consumers will not be justified in changing their views with respect to its detrimental characteristics.

No. 1983

BOILER-FURNACE REFRACTORIES

By E. B. Powell,[1] Boston, Mass.

Member of the Society

The advance in combustion equipment during the past twenty years has made continuously increasing demands upon boiler-furnace refractories. Improvement in refractories, however, has apparently not kept pace with these demands, and today various measures of wall cooling are being employed to overcome some of the limitations of refractories. The purpose of the paper is to promote interest in the further development of refractories. Some of the limitations of refractory raw materials, current practice and improvements at present under way in refractory manufacture, service conditions under which refractories are employed in boiler furnaces, characteristics of commercial refractories, and tests employed in analyzing these characteristics are discussed. Fireclay brick are given the greatest attention in this discussion as being the most universally used form of refractory.

There is need for further technical study both of boiler-furnace conditions and of refractory materials, and it is of special importance at the present that means be devised for interpreting laboratory test results in terms of service performance. The problem is complicated by the structural changes which refractories undergo in service. It is believed, however, that the microscope, which has been invaluable in the development of electrical porcelains and is now being directed upon refractories, will prove of very material assistance in this study.

THIS paper takes the form of a critical analysis, not for the purpose of criticism, but in order that it may be of value as a basis for constructive development.

2 The particular clays, finished brick, and slags to which reference is made in the paper are presented for the purpose of illustrating characteristics which may be found in the general class of

[1] Consulting Engineer, Stone & Webster, Inc.

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materials described; they are not intended as representative of the districts from which they are derived.

3 The author is greatly indebted to Prof. Gordon B. Wilkes, whose work is represented by data throughout the paper, and to Dr. Herbert Insley for his invaluable contribution of petrographic analyses and photomicrographs. He is also glad to express appreciation of the coöperation of G. A. Balz, M. F. Beecher, G. A. Bole, M. C. Booze, J. J. Brennan, R. F. Geller, J. E. Brinckerhoff, Dr. F. A. Harvey, C. F. Hirshfeld, A. M. Kohler, J. S. McDowell, F. H. Norton, R. C. Purdy, F. H. Riddle, R. A. Sherman, and Dr. R. B. Sosman; the representatives of the public-utility companies whose coöperation made possible the refractory surveys of 1922 and 1924; and the firebrick manufacturers whose frank discussion of their processes formed the basis of a large section of the paper.

General

4 A few years ago the comment was frequently heard that not only was the advance in combustion equipment and in the consequent requirements of the modern steam-boiler furnace outrunning the improvements in refractories, but was in fact checked by the limitations of refractories. Today various measures for cooling furnace walls by steam, water, and air are being employed to overcome some of these limitations. Yet, in these past few years, the development of refractories has by no means stood still, and further development will play an important part in the future of the boiler furnace.

5 It is the purpose of this paper to outline some of the factors in the service life of boiler-furnace refractories in the hope of furthering the movement for a clearer understanding of these factors so that better refractories and better utilization of them may ultimately result. The factors discussed deal chiefly with the composition and properties of materials, manufacture, properties of finished product, and service conditions. Tests are an essential aid to the technical analysis of the problem; therefore methods of tests are also examined.

6 The materials and manufacture of firebrick, their general properties, characteristics of failure, and the conditions to which they are subjected in service have been admirably described in the current technical press, and Circular 282 of the Bureau of Standards[1] gives a concise and excellent outline of the subject. Some of the essential features are reviewed on account of their direct bearing upon the service life of refractories and upon possibilities of refractory development.

[1] For this and other references, see bibliography, pp. 928-930.

CHARACTERISTICS OF MATERIALS—ALUMINA AND SILICA

7 The most important raw materials in the manufacture of boiler-furnace refractories, which come under the general classification of clays, are natural compounds and mixtures of silica and alumina in more or less hydrated form; probably next in importance are the diaspores and bauxites, hydrates of alumina. Although these materials are largely products of rock decomposition in the form of sedimentary deposits and are therefore contaminated with various impurities from other materials with which they have been associated, the essential constituents are silica and alumina. Accordingly, in considering the characteristics of boiler-furnace refractories, one is most concerned with the properties of these two oxides and with the effect upon their properties of combination between the two and of reactions with other materials which may be associated with them as natural impurities, or as additions in manufacture, or with which they may come in contact in service. A number of other materials used as refractories will also be discussed, but, with one exception, silicon carbide, these are all in the experimental class, and their commercial application to boiler-furnace requirements is still problematic.

8 Silica and alumina are believed to have combined in nature in at least three different molecular ratios: $Al_2O_3.2SiO_2$, the ratio of kaolin; $Al_2O_3.SiO_2$, the ratio of cyanite, andalusite, and sillimanite; and $3Al_2O_3.2SiO_2$, the ratio of mullite. It is possible that there are other natural compounds of silica and alumina. Clay [2] is a substance of rather indefinite composition in which silica and alumina are found in ratios varying from about 15 per cent alumina and 75 per cent silica in the siliceous clays of New Jersey and Colorado to about 50 per cent alumina and 35 per cent silica in the so-called nodular clays of Pennsylvania and burley clays of Missouri. By some investigators [3] these varying ratios are apparently regarded as resulting from a multiplicity of compounds and mixtures of these compounds. However, the only alumina-silica compound which seems to have been definitely identified as a significant constituent of natural clay is that of kaolin (and hydromica [4] which has practically the same proportions of alumina and silica as kaolin but in which some of the water is replaced by alkali), and clay is commonly regarded as kaolin with more or less excess of silica or of alumina, depending upon the deviation of the actual composition from that of the theoretical kaolin, $Al_2O_3.2SiO_2.2H_2O$, equivalent by weight to about 39.5 per cent alumina, 46.6 per cent silica, and 13.9 per cent water. On the other hand, recent investigations have shown that at high temperatures the kaolin compound is not stable, in fact, some investigators have held that clays break down to alumina, silica, and water at temperatures as low as 932 deg. fahr. (500 deg. cent.).

At somewhat higher temperatures, probably between 1652 deg. fahr. (900 deg. cent.) and 2192 deg. fahr. (1200 deg. cent.), alumina and silica recombine, but in an entirely different ratio.

9 The relations of silica and alumina at high temperatures are best shown by the equilibrium diagram of Bowen and Greig,[8] which is reproduced in Fig. 1. It will be noted that at the temperatures of the diagram there is but a single stable compound, $3Al_2O_3.2SiO_2$, which is known as mullite. This compound is perfectly homogeneous up to a temperature of about 3290 deg. fahr. (1810 deg. cent.) at which temperature it is shown as beginning to melt, breaking up into corundum, which is the crystalline form of alumina, and liquid. For mixtures containing less than the mullite ratio of alumina, the uncombined silica appears as cristobalite,

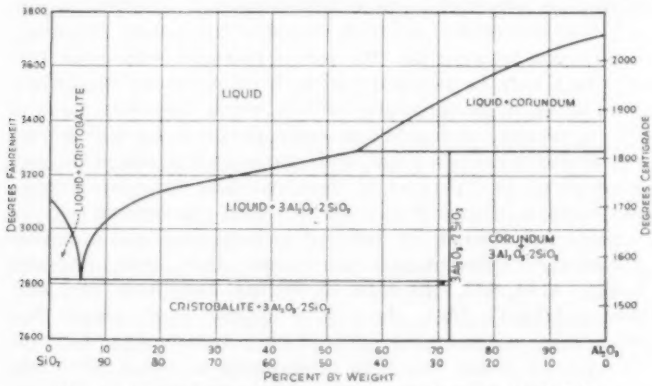

FIG. 1 EQUILIBRIUM DIAGRAM OF THE SYSTEM Al_2O_3-SiO_2
(Bowen and Greig)

one of the crystalline forms of that material, or as liquid. For mixtures of higher alumina ratio, mullite and corundum are shown below 3290 deg. fahr. (1810 deg. cent.) and corundum and liquid above this temperature.

EUTECTIC MELTING POINTS

10 The diagram presents the upper thermal limit of solidity for any possible mixture of materials in which alumina and silica are the major constituents. A single eutectic occurs, that shown between mullite and cristobalite, at the composition of about 5½ per cent Al_2O_3 and 94½ per cent SiO_2, with a melting point around 2813 deg. fahr. (1545 deg. cent.). From this diagram, it is apparent that any possible combination of alumina and silica below the mullite ratio must contain a liquid, more or less viscous, at all temperatures above 2813 deg. fahr. (1545 deg. cent.). In fact, it is only the pure minerals which have so high a eutectic melting point as that shown on this diagram. While it is true that not

all possible impurities have been investigated in their effect upon alumina-silica mixtures, extensive studies have been made of effects of lime and magnesia, and some work has been done on effects of iron compounds and of alkalis. All of these studies point to the formation of a series of eutectics with progressively lower melting temperatures as the number and quantity of impurities are increased.

11 As an illustration of eutectic formations: The separate minerals magnesia (MgO) and lime (CaO) have fusion temperatures of about 5072 deg. fahr. (2800 deg. cent.) and 4658 deg. fahr. (2570 deg. cent.), but combine to form a eutectic having a melting point of 4172 deg. fahr. (2300 deg. cent.); alumina, having a

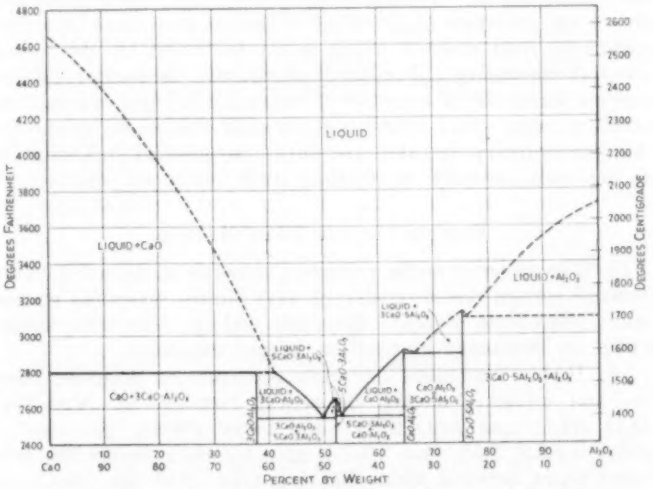

FIG. 2 EQUILIBRIUM DIAGRAM OF THE SYSTEM CaO-Al_2O_3
(Rankin and Wright)

melting point of 3722 deg. fahr. (2050 deg. cent.), and silica, having a melting point of 3110 deg. fahr. (1710 deg. cent.), the two minerals melting in eutectic mixture at 2813 deg. fahr. (1545 deg. cent.), together combine with magnesia to form a eutectic having a melting point of 2453 deg. fahr. (1345 deg. cent.) and with lime to form a eutectic having a melting point of 2138 deg. fahr. (1170 deg. cent.). In these eutectics the magnesia and lime are in fairly large proportions, the mixtures being respectively 18.3 per cent Al_2O_3, 61.4 per cent SiO_2, and 20.3 per cent MgO in the first and 14.75 per cent Al_2O_3, 62 per cent SiO_2, and 23.25 per cent CaO in the second. Where impurities are relatively small in amount and very intimately mixed with the fundamental constituents, eutectics of higher melting points may be formed as shown for lime and alumina on the equilibrium diagram* of Fig. 2. On

the other hand, where the impurities occur in concentrated particles, especially if in the mixture of alumina and silica the silica is present in higher than the normal kaolin ratio, the eutectic of lowest melting temperature is formed at that point.

12 For mixtures involving only two minerals, softening will, in general, occur very rapidly as the eutectic melting point is reached. Where there are several minerals in the mixture, however, time is required for the eutectic formations, and melting takes place progressively. This is especially true if there are concentrations of materials of different fusion temperatures and different physical characteristics, in which case the material of lowest softening temperature may melt and then gradually dissolve the material of higher softening temperature. Also, under some conditions the structure of the purer materials may serve as a comparatively rigid skeleton which is not destroyed by the solvent action of the eutectic. It is for these reasons, the partial and progressive formation of eutectics of comparatively low melting temperature while other constituents remain practically unaffected, that the ordinary firebrick are subject to gradual softening and do not melt abruptly on reaching their fusion temperature.

Volume Change from Heating

13 The majority of the refractory minerals in the natural state undergo certain non-reversible or very slowly reversible changes under the effects of heat; they also undergo reversible changes which are commonly grouped as thermal expansion.

14 Highly aluminous materials occurring as diaspore, bauxite, and gibbsite, having theoretical formulas of $Al_2O_3.H_2O$, $Al_2O_3.2H_2O$, and $Al_2O_3.3H_2O$, and initial specific gravities[7] of about 3.3-3.5, 2.55, and 2.3-2.4, give up their chemically combined water between about 590 deg. fahr. (310 deg. cent.) and 1004 deg. fahr. (540 deg. cent.) and at higher temperatures, around 1958 deg. fahr. (1070 deg. cent.) to 2300 deg. fahr. (1260 deg. cent.), are converted to corundum, the crystalline form of alumina, a material of much higher density. Shrinkage continues, however, and is only completed by actual fusion, when the specific gravity becomes about 3.9. These changes of density are non-reversible. The reversible or thermal expansion of the final fused alumina is practically constant, the coefficient of expansion[8] between 77 and 1652 deg. fahr. (25 and 900 deg. cent.) being 0.000004 (0.000007 per deg. cent.).

15 Natural silica is free from chemically combined water but on heating undergoes several transformations, some reversible, and others so slowly reversible as to be practically non-reversible under ordinary conditions. Fenner[9] has identified seven distinct modifications of silica, the three principal ones being quartz, tridymite, and crystobalite, all possessing alpha and beta phases. As presented

in his paper, The Stability Relations of the Silica Minerals, the inversion temperatures for the principal forms are:

$$1598 \pm 18 \text{ deg. fahr. } (870 \pm 10 \text{ deg. cent.})$$
$$\text{quartz} \rightleftharpoons \text{tridymite}$$
$$2678 \pm 18 \text{ deg. fahr. } (1470 \pm 10 \text{ deg. cent.})$$
$$\text{tridymite} \rightleftharpoons \text{crystobalite}$$

The volumetric relations of these minerals as presented by McDowell [10] are shown in Fig. 3. The specific gravity of quartz is 2.650; of cristobalite, 2.333; of tridymite, 2.270; and of quartz glass, 2.210. Accordingly, the complete transformation of quartz to cristobalite gives a volume increase of 13.6 per cent and that of

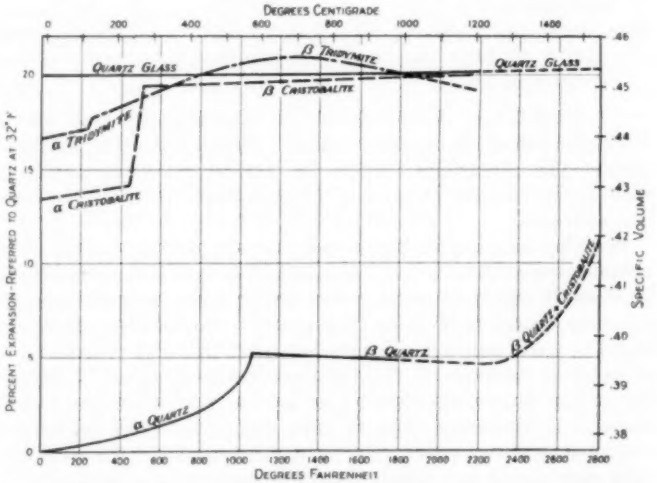

FIG. 3 VOLUME-TEMPERATURE CHARACTERISTICS OF SILICA MINERALS

quartz to tridymite, 16.8 per cent. While these changes are reversible, the reversion under normal conditions is very slow. In general, the transformation from quartz to tridymite, except possibly in the presence of a high percentage of flux, is not direct, but cristobalite is first formed and, at temperatures below 2678 deg. fahr. (1470 deg. cent.) and above 1598 deg. fahr. (870 deg. cent.), there occurs a gradual inversion to tridymite. Below 2372 deg. fahr. (1300 deg. cent.), the change from quartz to cristobalite is extremely slow, but at 2678 deg. fahr. (1470 deg. cent.), quartz begins to melt and in melting is converted almost immediately into cristobalite. Above this temperature the change is fairly rapid, taking place in a few hours. At 3110 deg. fahr. (1710 deg. cent.) cristobalite and at 3038 deg. fahr. (1670 deg. cent.) tridymite melt to quartz glass which is stable at atmospheric temperatures and at tempera-

tures above the fusion points of cristobalite and tridymite but, under prolonged heating at lower temperatures, above 1598 deg. fahr. (870 deg. cent.), is subject to the same processes of inversion as the original quartz.

16 According to Fenner[9] the inversions of the alpha and beta phases of the silica minerals occur as follows:

Alpha quartz→beta quartz 1067 deg. fahr. (575 deg. cent.)
Beta quartz→alpha quartz 1058 deg. fahr. (570 deg. cent.)
Alpha tridymite→beta$_1$ tridymite * 243 deg. fahr. (117 deg. cent.)
Beta$_1$ tridymite→beta$_2$ tridymite * 325 deg. fahr. (163 deg. cent.)
Alpha crystobalite→beta crystobalite 526.3 to 427.5 deg. fahr. (274.6 to 219.7 deg. cent.), depending upon previous heat treatment.
Beta crystobalite→alpha crystobalite 464.9 to 388.6 deg. fahr. (240.5 to 198.1 deg. cent.), depending upon previous heat treatment.

17 At ordinary temperatures the silica minerals exist only in the alpha phase, changing to the beta phase as the temperature is raised. The alpha-beta changes are rapidly reversible and occur with every heating or cooling through their range of inversion temperatures.

18 The diagram of Fig. 3 indicates the general nature of the volume changes which occur in connection with the alpha-beta inversions. While there seems to be some question as to the absolute accuracy of the work of Le Chatelier[11] on which these curves are based in part, they are generally accepted as fairly representative of relative volumes of the different forms of silica. Tridymite shows the lowest thermal expansion, in fact there is a slight reversal of expansion beyond 1292 deg. fahr. (700 deg. cent.). Cristobalite shows practically no expansion above 527 deg. fahr. (275 deg. cent.) but below that temperature, accompanying the alpha-beta inversions, there are sharp changes of volume. Quartz glass is very nearly constant in volume throughout its solid temperature range.

19 The natural alumina-silica mixtures of the fireclays include more or less chemically combined water and are subject to non-reversible volume changes on heating, accompanying expulsion of the combined water and condensation of structure with vitrification, which are governed by the rate of heating, the ratio of these two minerals, and the extent and nature of impurities contained. Where the alumina-silica ratio is approximately that of kaolin or higher, the reversible expansion, while varying widely with different clays, seems to give a practically straight-line relation between temperature and volume. For higher percentages of silica, especially where the silica is equal to 70 or 75 per cent of the mixture, the

* Reversions on cooling not very sharp.

effect of the alpha-beta inversions of quartz and crystobalite appear, giving a broken expansion curve. These irregularities of expansion may be smoothed out by conversion to tridymite.

20 The non-reversible and reversible volume changes in clays are discussed at greater length in connection with the burning and service characteristics of the molded brick.

MANUFACTURE OF FIRECLAY BRICK

Raw Materials: Mining and Inspection

21 The refractory clays used in firebrick manufacture occur in strata or seams of a wide range in thickness, from about 2 to 20 ft. or more, extending for miles, and sometimes to depths of several hundred feet under the hills; or in lenses or basin-shaped deposits of from a few hundred square yards to several hundred acres in area and 12 to 40 ft. or more in depth. Frequently clays of different character and distinguishable by color or texture lie one above the other, and there are concentrations of impurities sometimes at spots in the seam but generally at the boundaries above and below the seam. Sometimes such impurities are scattered in small particles all through the seam, and in other seams continuous veins of highly impure clay are encountered.

22 The first step in checking the quality of material to be used in the brick is made when the clay bed is surveyed. With a number of manufacturers this survey takes the form of thorough prospecting; drill cores are taken at close intervals over the property and from the tests and analyses made on these the composition and disposition of the clays in the seam are fairly definitely known well in advance of actual mining. Some clay seams such as those of New Jersey, and lens deposits such as those of kaolin and diaspore, are comparatively near the surface so that they may be stripped of overburden including impurities of the upper section of the deposit and exposed to visual inspection.

23 In the general mining operation the clay is broken out of the seam by light charges of dynamite and loaded into cars by hand; in some of the open-pit workings, however, steam shovels are used. It is the duty of the miner to load the different grades of clay into separate cars and to discard impurities, this being done under the inspection of the mine foreman. The most troublesome of the separable impurities are the iron compounds, carbonaceous materials, and sand. It is generally the duty of the mine superintendent to visit the workings daily. In some mines, inspectors make daily visits to all parts of the working face where clay is being mined; in some, ceramists make visits once or twice weekly or every two weeks. In many instances a second inspection is made of the clay as it is unloaded at the plant, and in some of these the cars bear the identification of the miner, who is penalized

for impurities detected in his material as delivered to the plant. In some plants, daily samples are taken of the raw materials, and fusion tests, sometimes supplemented by tests for iron content, are made weekly on the composite sample; in others, fusion tests and analysis of raw materials are made monthly; and, in still others, tests are made only when material of unusual appearance has been uncovered.

24 The majority of manufacturers make a practice of producing two or more grades of brick in order to provide for disposal of their inferior clays and also to maintain a balance of mining and utilization for the different grades of clay which the nature of the deposit may require to be moved. Up to within a very few years, the uniformity or the lack of it in the quality of clay entering the first-quality brick was not uncommonly dependent rather largely upon the miner alone and selection of material based upon appearance, supplemented by tests of rather simple sort such as burning of specimen brick in a kiln with others of supposedly standard quality. Not only was much inferior clay accepted under this system, but some of the very high-grade clays were discarded. Many manufacturers report marked improvement in quality of product due to more careful and more intelligent sorting of clays which has been put into effect within the past two years.

Classification of Fireclays

25 Fireclays may be classified broadly according to physical properties as flint and plastic. There are, however, clays of intermediate properties and known as semi-flint and semi-plastic. The plastic clays are of the softer structure and break down readily on exposure to the weather; they possess the property of developing binding power when mixed or tempered with water. In chemical composition the plastic clays tend to a lower ratio of alumina to silica than the flint clays and, as a matter of geological coincidence, they generally contain a higher percentage of impurities and have a lower fusion point.* The flint clays are of much harder structure, and in general are but slowly affected by the weather. They usually possess initially practically none of the binding qualities of the plastic clays, but in some of the flint clays a degree of plasticity can be developed by prolonged weathering or by continued grinding and tempering. The semi-flint and semi-plastic clays possess intermediate binding qualities, and frequently have a high degree of refractoriness approaching that of the flint clays.

26 In Table 1 are given chemical analyses and approximate fusion temperatures for a number of clays used in firebrick manu-

* There are, of course, notable exceptions in both alumina-silica ratio and degree of purity. Some of the higher-grade plastic clays are very low in impurities.

factured in different parts of the country. These analyses are not intended as necessarily typical of the different districts from which they are derived, but are presented as illustrating some of the characteristics of clays dealt with in firebrick manufacture. The analyses as given refer to the dried clay. The free moisture in clays from underground mines will average around 1 to 2 per cent, with total variations of about 2 per cent; in beds nearer the surface the moisture will frequently be higher, and on account of the influence

TABLE 1 ANALYSIS OF CLAYS

FLINT

	Missouri	Pennsylvania	Maryland	Ohio	Kentucky
Ignition	13.78	13.63	7.90	13.50	13.60
Silica	42.70	43.10	50.27	43.52	41.00
Alumina	39.72	39.40	39.40	37.86	37.22
Iron oxide	1.14	1.35	1.86	1.79	5.78
Titania	2.22	2.01	....	1.51	2.00
Lime	0.13	0.13	0.36	0.44	0.34
Magnesia	0.29	0.08	0.01	0.66	1.01
Alkalis	0.22	0.20	0.53	0.80	1.57
Fusion, cone	34-35	33-34	33-34	33	32
deg. fahr.	3182	3146	3146	3128	3101
deg. cent.	1750	1730	1730	1720	1705

SEMI-FLINT AND SEMI-PLASTIC *

	Missouri	Pennsylvania	New Jersey	Kentucky	Colorado
Ignition	12.66	11.53	12.38	14.08	14.16
Silica	49.64	46.20	51.04	43.46	49.02
Alumina	33.69	36.33	32.31	37.45	34.78
Iron oxide	1.30	1.65	1.29	1.19	0.72
Titania	1.56	2.16	1.92	2.80	0.96
Lime	0.31	0.18	0.37	0.30	0.39
Magnesia	0.43	0.27	0.34	0.64	0.24
Alkalis	0.37	1.72	0.12	0.93	0.37
Fusion, cone	32-33	32	32-33	32-33	32
deg. fahr.	3115	3101	3115	3115	3101
deg. cent.	1713	1705	1713	1713	1705

PLASTIC

	Missouri	Pennsylvania	Maryland	New Jersey	Kentucky
Ignition	12.10	12.68	9.15	5.60	9.66
Silica	56.76	44.60	50.80	76.24	55.92
Alumina	26.57	37.35	37.37	15.76	24.71
Iron oxide	1.60	1.17	0.63	0.52	4.79
Titania	1.75	2.16	....	1.68	1.52
Lime	0.44	0.30	0.08	0.26	0.34
Magnesia	0.17	0.42	0.14	0.29	1.01
Alkalis	0.10	1.57	1.80	0.23	1.57
Fusion, cone	30	32-33	31-32	30	20-26
deg. fahr.	3038	3115	3083	3038	2849
deg. cent.	1670	1713	1695	1670	1565

	Missouri	A Maryland Seam			
	Burley	No. 1	No. 2	No. 3	No. 4
Ignition	13.30	11.36	8.68	7.32	1.22
Silica	34.62	46.68	57.34	61.40	94.80
Alumina	48.03	36.32	28.55	26.28	3.40
Iron oxide	1.09	2.39	1.58	1.16	0.22
Titania	2.28	2.01	3.45	1.44	0.52
Lime	0.25	0.07	0.17	0.13	0.11
Magnesia	0.56	0.46	0.13	0.54	0.15
Alkalis	0.44	0.32	0.71	1.13	0.35
Fusion, cone	Above 33	32-33	28-29	29	32
deg. fahr.	Above 3128	3115	2990	3002	3101
deg. cent.	Above 1720	1713	1643	1650	1705

* Regarded as semi-flint in the New Jersey district, but the characteristics of this clay would classify it as plastic in any other district.

of the weather, the amount will be more variable. The ignition loss, though including some organic matter, is chiefly made up of water chemically combined with the mineral constituents. In general the chemically combined water will vary roughly with the ratio of alumina to silica, as water is found chemically combined in all natural aluminous materials and is entirely absent in quartz.

27 The ratio of alumina and silica affords only a general index to the character of the clay. In fact, clays from different parts of the country or even from closely adjoining seams may be of almost identical chemical analysis and yet possess markedly different characteristics in both natural and finally burned states. This difference is due partly to small differences in impurities contained; some investigators hold that it is also due to differences in combination of

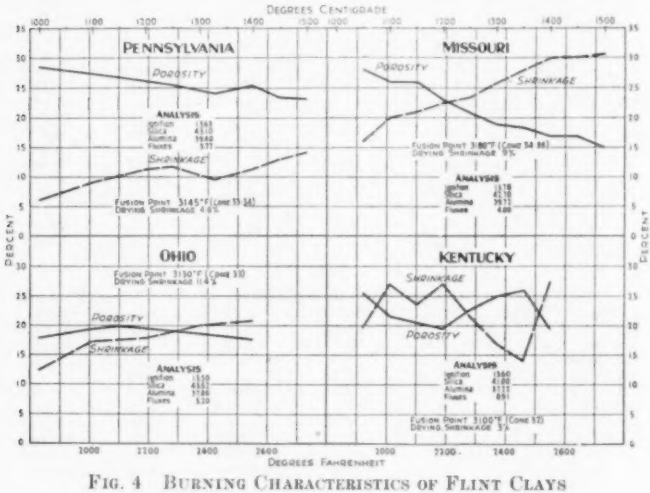

FIG. 4 BURNING CHARACTERISTICS OF FLINT CLAYS

alumina and silica. Analyses of the high-grade Missouri, Pennsylvania, Kentucky, and Ohio flint clays show but slight variation in alumina-silica ratio and little difference in total impurity content. That they are different in service characteristics is generally recognized. Another illustration is given in the first two of the Maryland clays, shown in Table 1, which have almost identical alumina-silica ratio and about the same total impurities, yet, in physical characteristics, one is a flint clay and the other plastic.

28 An illustration of diversity in clays is given by the series of Maryland clays, the analyses for which are numbered 1, 2, 3, 4, for the top, second, third, and bottom strata in the same seam. These analyses represent four distinct clays lying one above the other in strata 3 to 4 ft. thick in the same seam, the silica content ranging from about 47 to 95 per cent.

29 The fusion temperatures for clays at the opposite extremes of alumina-silica ratio show a reasonably consistent relation but at the intermediate ratios the effect of variations in basic oxides and alkalis will generally mask the effect of alumina-silica variations.

30 Titania is frequently not separately determined but included with the alumina, although it is generally regarded as an impurity contributing to the lowering of the fusion point as iron, lime, magnesia, and alkalis.

Shrinkage and Porosity Changes

31 Figs. 4 and 5 show shrinkage and porosity characteristics of some of these clays,[12] the approximate composition for conveni-

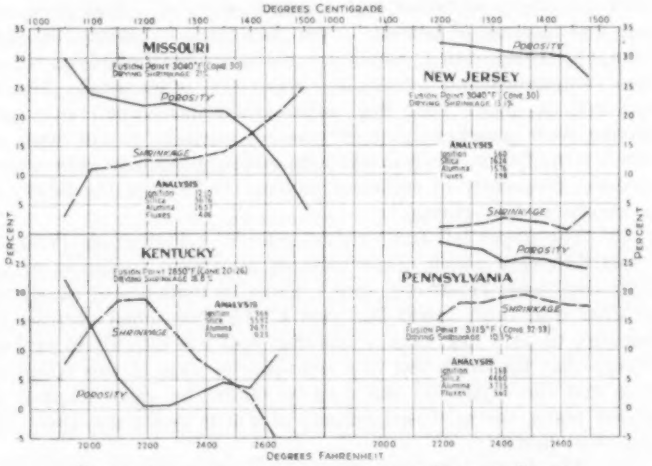

Fig. 5 Burning Characteristics of Plastic Clays

ence being shown beside the curve. The burning shrinkage values are based upon the dried brick and, as will be seen from the data on the charts, are preceded by drying shrinkages which vary widely, from 3 to more than 20 per cent. As the temperature is raised above 212 deg. fahr. (100 deg. cent.) to about 450 or 500 deg. fahr. (232 or 260 deg. cent.) more of the water held mechanically is driven off. The chemically combined water is but slightly affected until higher temperatures — above 900 deg. fahr. (482 deg. cent.) — are reached. Between 900 deg. fahr. (482 deg. cent.) and about 1650 deg. fahr. (899 deg. cent.) the kaolin molecule of the clay is completely broken down, and at about this point the average clay shows its greatest porosity. Between 1450 deg. fahr. (788 deg. cent.) and 1700 deg. fahr. (927 deg. cent.) a structural change occurs, which is regarded by some as due to the formation of mul-

lite and by others to molecular rearrangement of alumina, beyond which the average clay begins to contract and porosity decreases. As these volume and porosity changes are only made possible by the softening of the clay, the temperature at which they begin and the final rate and extent of change for any given temperature are very largely dependent upon the amount and nature of impurities contained in the clay.

32 The upper right-hand curves of Fig. 4 represent a high-grade Missouri flint. The shrinkage and change in porosity occur with reasonable uniformity up to a temperature of about 2552 deg. fahr. (1400 deg. cent.) when apparently, although there is further decrease of porosity, the volume remains practically constant to 2732 deg. fahr. (1500 deg. cent.). The set of curves immediately below refer to a Kentucky flint clay which is included to show the effects of rather high impurity content. Marked expansion and increase of porosity occur, beginning at about 2192 deg. fahr. (1200 deg. cent.), which are probably due to breakdown of iron compounds and release of oxygen and other gases. This is followed at about 2462 deg. fahr. (1350 deg. cent.) by rapid fluxing with shrinkage and decrease of porosity to the limit of the determination. Such marked expansion introduces very severe stresses, frequently of sufficient violence to rupture the brick; it is known as "overfiring."

33 The upper left-hand curves refer to a high-grade Pennsylvania flint clay which, while showing expansion beginning about 2282 deg. fahr. (1250 deg. cent.), gives no evidence of overfiring at 2732 deg. fahr. (1500 deg. cent.). The continued decrease of porosity to 2417 deg. fahr. (1325 deg. cent.) is difficult to explain except that the brick was in a semi-plastic state at this temperature and the expansion was largely caused by gases trapped in sealed pores, the increased volume of which would not affect the apparent porosity.

34 The lower left-hand curves show the characteristics of a flint clay from Ohio. The shrinkage of this clay continues at fairly uniform rate to about 2372 deg. fahr. (1300 deg. cent.) beyond which the volume changes but slightly up to 2732 deg. fahr. (1500 deg. cent.), the limit of the determination. The porosity increases continuously to about 2102 deg. fahr. (1150 deg. cent.), probably due to gradual release of gases during this period, and shows but slight decrease to 2552 deg. fahr. (1400 deg. cent.).

35 The curves of Fig. 5 are for plastic clays. The upper right-hand curves refer to one of the highly siliceous clays of New Jersey. This clay shows very small volume shrinkage and but slight change in porosity to about 2624 deg. fahr. (1440 deg. cent.) when apparently final contraction begins. The expansion from about 2408 deg. fahr. (1320 deg. cent.) to 2624 deg. fahr. (1440 deg. cent.) is probably due to some molecular change or to changes in crystalline structure. The next set of curves below are for a Pennsylvania

plastic clay which within the range of values determined shows only very gradual changes of volume and porosity which compare favorably with some of the highest-grade flint clays. The upper left-hand curves are for one of the high-grade Missouri clays and show progressive shrinkage accompanied by corresponding decrease of porosity to 2732 deg. fahr. (1500 deg. cent.). As compared with the Missouri flint clay the change of volume and porosity occurs very rapidly beyond about 2462 deg. fahr. (1350 deg. cent.). The lower curves at the left are from another of the lower grades of Kentucky clay. This clay overfires so badly as to expand beyond the initial volume at about 2579 deg. fahr. (1415 deg. cent.).

36 It would be expected that the drying and burning shrinkages of different clays would bear some relation to the alumina-silica content. On account of the influence of other factors, however, there seems to be no such definite relation between clays from different seams. On the other hand, for the same clay, variation in silica content, which in that case is largely in the nature of variation in amount of uncombined silica or sand contained in the clay as an impurity, is reflected in shrinkage characteristics and, in some clays, constitutes one of the most serious problems in maintaining dimensions of finished material.

General Processes of Manufacture

37 For the purpose of discussion the manufacture of fireclay brick from the raw materials may be considered under four operations: preparation, molding, drying, and burning. The methods employed under these different operations vary with the character of the clay, the general process of manufacture, the type of refractory material being made, and the individuality of the manufacturer and equipment of his plant. Broadly speaking, there are three general processes of refractory manufacture: the hand-made process, and two machine-made processes; one of the latter being a wet process variously known as steam-press, power-press, and stiff-mud, and the other, the dry-press process. The characteristics distinguishing these processes are chiefly in the consistency of the clays and in the methods of molding. Fig. 6 shows, diagrammatically, arrangements of equipment and sequence of operations typical of the three processes.

Preparation

38 Preparation consists essentially of proportioning of the materials, and grinding and kneading with the addition of water, to effect thorough mixing of materials and to develop the plasticity of the clay. Plasticity is essential to give bonding, toughness, and strength to the unburned brick. The molded brick must possess sufficient strength to stand handling during the drying process and to support the weight of other brick and shapes without distortion

when piled to a height of 10 ft. or more within the kiln. Many of the hard and most refractory clays possess insufficient plastic qualities, and soft clays are added to permit development of the necessary cohesive strength.

39 In this country a comparatively small percentage of clay for firebrick manufacture is weathered before use. Some clays,

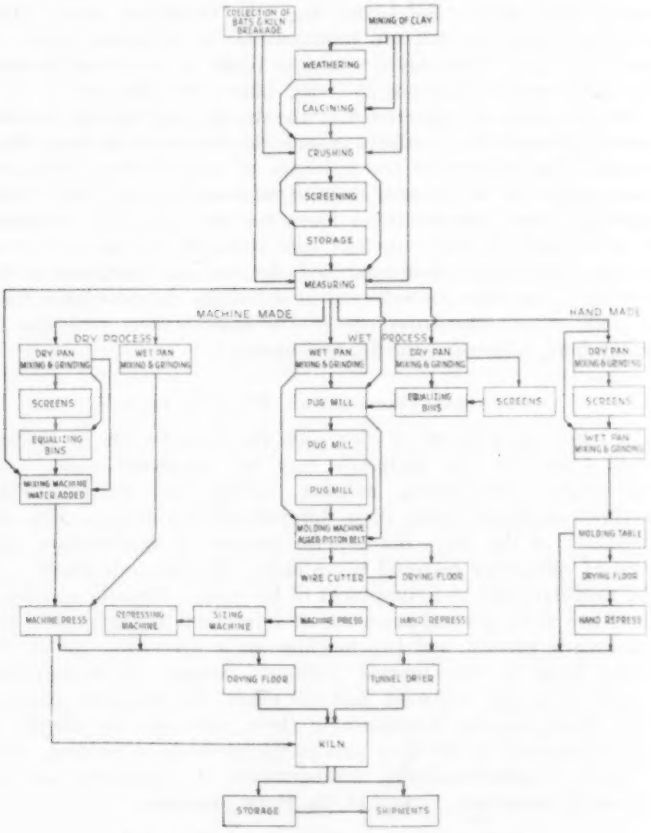

FIG. 6 MANUFACTURE OF FIRECLAY BRICK

apparently, are not improved by weathering. Probably in the majority of clays, weathering, by promoting oxidation and organic development with mechanical disruption of the structure, improves the plasticity. Thus, weathering may permit the manufacture of a higher-grade brick by avoiding the necessity for adding a less refractory plastic clay. With some clays, part of the impurities may be washed out by weathering, although the actual improve-

ment so effected is probably very small; surface impurities are made more apparent by oxidation. Weathering for firebrick manufacture, however, is chiefly employed as a means of breaking up the larger pieces of clay and so avoiding the necessity for a preliminary crushing operation.

40 Most fireclays shrink in drying and burning; some of them when molded into bricks have linear shrinkage in drying as high as 10 per cent, and further shrinkage almost equally high in burning. Such shrinkage would put the structure of the material to very severe stress. Many manufacturers add calcined, or burned, material in mixing their clays as a measure of reducing the shrinkage and otherwise improving the quality of the product. Calcined material seems first to have been used in the manufacture of large shapes. Such bodies frequently have linear dimensions of more than three feet, and any reduction in shrinkage is of great assistance toward maintaining the homogeneity of the mass. At some plants the calcined material consists merely of waste brick, warped or broken in manufacture, and frequently the purpose is merely to afford a means for disposal of kiln breakage rather than to improve the quality of the product. At a number of plants, however, the calcined material is obtained by burning the raw clay.

41 There are various methods of calcining clay. The plastic clays for this purpose are usually molded into bats of somewhat larger dimensions than standard brick and burnt in regular down-draft kilns. One of the manufacturers has devised a special type of continuous kiln in which the plastic clays are burned without previous molding. With the harder clays it is frequently the practice to select the larger lumps for calcining. These are piled in regular kilns of the down-draft type and burned as standard brick. Other manufacturers use kilns of the up-draft type for calcining, a number using the continuous, vertical-shaft type of kiln.

42 In calcining clays, the kilns are frequently fired much more rapidly than would be practicable with the molded brick, and rarely is any attempt made to measure accurately the temperature attained in the kiln. A few manufacturers, however, use the full equipment of recording and optical pyrometers, following a definite time-temperature schedule in calcining their clays. Others set cones at different points in the pile and determine the burn by the softening of these cones. The majority judge the calcine kiln temperatures only by eye. In practically all plants the extent of the calcine is finally gaged or checked by inspection of samples which are withdrawn toward the end of the burn.

43 In general the raw clays as received at the plant and the calcined materials which are used are first separately crushed in jaw, rotary, or other types of coarse crushers and stored in separate bins. This preliminary crushing will frequently reduce the material only sufficiently to permit satisfactory handling in chutes.

44 With the majority of manufacturers the proportioning of materials is done by volumetric measurement. Where the processes are of intermittent type, the measuring wagon, or buggy, fitted with adjustable partitions and having a total capacity equivalent to that of the mill in which the mixing is done, is commonly used. For continuous processes some of the manufacturers use measuring wheels or special proportioning feeders. With a few manufacturers the materials are proportioned by actual weight. In many instances, however, the measuring apparatus takes the simpler form of shovel and wheelbarrow.

Hand-Made Process

45 In the hand-made process of manufacture the clays and calcined materials are usually ground and mixed together in what is known as a "wet pan." This is a type of mill shaped like a circular pan with sides about 10 to 14 in. in height and about 6 to 9 ft. in diameter with revolving bottom on which bear two heavy steel-tired mill wheels 4 or 5 ft. in diameter mounted at opposite ends of a horizontal shaft which floats in vertical guides. These rim-running mill wheels, or "mullers" as they are called, crush and knead the material as it is carried under them. Scrapers fixed above the bottom of the pan agitate the mass and constantly return it to the path of the wheels. Frequently the crude materials are delivered to this machine without previous crushing. Water is added and the grinding and mixing operation is continued until the desired plasticity is secured. Sometimes the plastic clay is ground to a paste before the flint clay and calcined material are fed to the mill.

46 The control of the water supply to the wet pan in preparing clays for hand molding is invariably left to the judgment of the operator. The time of mixing, however, is sometimes definitely scheduled and this schedule departed from only when due to some unusual condition the desired plasticity has not been secured. When the mass is in a satisfactory condition the pan is unloaded by means of an automatic emptying device which commonly takes the form of a plow which is lowered at one side of the pan. Occasionally, where it is necessary to elevate the clay, the plow delivers the batch on to a small belt conveyor.

47 Continuing the hand-made process, from the wet pan the clay is carried to the molding table where it may be inspected by the molding foreman for plasticity. If the clay is considered satisfactory the molder cuts it into lumps, one at a time, which he quickly squares by bumping a few times on his table, then raises high above his head and throws with a strong downward motion into the previously sanded mold. The surplus material is cut off sharply, even with the edge of the molding frame. The unused material containing some sand is returned to the original

mass of clay. In the more careful plants the cut face of the returned clay is pressed against the face of the unused clay so as to prevent lamination from the sand. The molding frames hold four or five molds and are generally made of wood. After filling they are carried to the drying room and dumped face down onto the drying floor. After preliminary drying, usually three or four hours, during which they are sometimes set up on a 9 by 2½-in. edge, the brick are repressed by hand and returned to the drying floor for final drying; at this time they are generally stacked 4 to 7 high with a space of an inch or two all around for ventilation.

48 In the hand-made process the force with which the tempered material is thrown into the mold is necessarily subject to wide variation as is also the pressure applied in finally repressing the brick. Dependence on the human factor in preparation, molding, and repressing has necessarily tended against uniformity of product as made by this process. Some manufacturers avoid a part of this variable factor by molding in a special machine in which the clay, prepared as for hand molding, is forced into molding frames under comparatively light pressure and cut off automatically in a manner very similar to that used in hand molding. The brick are subsequently repressed by hand as in the case of the ordinary hand-made brick. With this machine molding, the risk of working sand into the clay is practically eliminated.

Machine-Made Wet Process

49 In the machine-made wet process the crushing, grinding, and mixing instead of being conducted in a single machine are more commonly carried on in separate machines. In usual practice the crude materials are crushed and ground in a dry pan which is very similar to a wet pan except that for this purpose it is provided with a slotted bottom through which the crushed materials are discharged continuously as ground. In some plants the several materials are crushed separately and sized through screens, generally of the rotating or vibrating type, stored in separate bins, and then proportioned according to size and material as fed to the mixing machine.

50 Some manufacturers use special precautions in bins to prevent segregation of sizes in their crushed materials, a number of them reporting that considerable difficulty had been encountered from variation in sizing before such measures were adopted. Where the crude materials are crushed separately before mixing they are sometimes mixed in batches and ground together in a dry pan. In other plants, however, the mixing takes place in a wet pan, water being added as the materials are mixed and ground together.

51 In some plants, the mixing process for machine-made brick is completed in the wet pan, following the common practice for

hand-made brick, but in the majority of plants the various crude materials either with or without previous mixing are delivered to a pug mill, which consists essentially of a mixing trough in which are mounted one or two lengthwise horizontal shafts carrying suitable paddles and blades set at an angle to give the mixture a slight forward movement. In the pug mill the materials are mechanically mixed together and tempered with water which, in the majority of plants is hand-controlled and left entirely to the judgment of the operator. In a few plants the supply of water to the pug mills is governed by an automatic proportioning device known as a "poidometer," which consists essentially of a counterbalanced conveying belt over which the clay charge is fed continuously; the variations in weight of the clay, being reflected in a rising or falling of the belt, serve to check and open the water-supply valve. Even in plants equipped with the poidometer, however, there is frequently supplemental hand adjustment of water supply to secure the requisite consistency of the clay.

52 In many plants the pugging is done in two or three mills which discharge one to another in series. The final pug mill discharges into the hopper of the molding machine, which is the more commonly used auger type of machine, or the piston type of machine which in some plants has succeeded the auger machine, or a machine of the belt type. In the first two machines a column of the tempered clay is extruded through a die and conveyed by belt to the wire cutting machine. In the belt type of machine the die follows the clay to the wire cutting machine. Unless the clays are thoroughly mixed and of the proper plasticity, the auger type of machine is likely to leave laminations in the brick. The development of the piston-type machine, and of the belt-type machine, have resulted from efforts to avoid risk of these laminations. The wire-cut blanks are fed by hand to the machine press, in which they are given the finished shape. After pressing, the molded brick are conveyed on pallets to the drying floor or stacked on cars for drying in a tunnel drier.

Machine-Made Dry Process

53 For the dry-press process of manufacture, the materials are generally separately crushed, screened, and stored in bins as in the case of the machine-made brick using the wet process. From the bins the material is conveyed in the desired proportions to dry pans to be ground and mixed together, if the materials have not been previously ground. From the dry pan the ground materials pass to a mixing machine where just sufficient water is added to give a slight cohesiveness or stickiness to the mass. In some plants the grinding and mixing and addition of moisture required are done by wet pan and the amount of water added definitely deter-

mined in advance. In either case the damp mixture is delivered to the press as a but slightly moistened powder and pressed into shape under high pressure, 2000 to 4000 lb. per sq. in. These dry-press brick contain only about 7 to 8 per cent moisture as molded, and are frequently conveyed direct from the press to the kiln without preliminary drying.

Special Processes

54 There are other special processes, very similar to the dry-press process, by which materials in a somewhat more moist state are molded into brick. There are also processes by which the column of tempered clay is extruded downward as a vertical beam into the die and wire-cut. This latter process has possibilities of much nearer approach to the hand-made brick, as a much softer, more plastic mixture may be used than with the more usual type of auger machine, which extrudes the clay as a horizontal beam.

55 Two of the manufacturers are now using a special process for sizing machine-made brick in which the extruded column of clay is conveyed by belt to a rotary wire cutter where it is cut into blanks which are in turn fed by hand to a power press, after which process the brick are immediately shaved to predetermined thickness and repressed, when they are ready for drying. The users of this process claim an exceptional uniformity of size for their brick.

Limitations in Sizing

56 Under any process of manufacture the molds in which the brick are finally pressed are of steel and subject to constant wear, tending to a gradual increase in size of product. These molds must be adjusted to some mean water content, composition, and sizing of mixture. Variations in water content directly affect both drying shrinkage and final porosity of the product. Variations in composition, especially variations in amount of calcined material or of sand included as impurity or "free" silica, and in sizing of mixture affect both drying and burning shrinkage. When grind and water content are governed by the judgment of operators it would seem hardly possible to avoid variations in size of product.

Drying

57 Drying is done either on dry floors in relatively large rooms with natural circulation of air, supplemented at times by artificial circulation, or in enclosed driers of the tunnel type in which circulation is largely artificial. The drying floors are of wood, steel, stone, or concrete. Wooden floors are in general heated only by the warmth of the room below or by circulation of warm air. Concrete, steel, and stone floors are heated by live or exhaust steam in ducts beneath them; in some plants waste heat from the kilns

is used. These floors are generally divided into a number of sections and the temperature of the different sections controlled at the discretion of the foreman in accordance with weather conditions, the type of ware to be dried, and the stage of drying, and may vary from 60 or 80 deg. fahr. (16 or 27 deg. cent.) to very nearly 200 deg. fahr. (93 deg. cent.).

58 Warm-air circulation where provided is supplied from a system of overhead ducts. The air for this purpose is generally heated by steam coils. Some manufacturers, however, circulate waste heat from cooling kilns during the night and at off hours when brickmaking is not in progress.

59 The tunnel-type driers are in general heated either by circulation of waste heat from the cooling kilns or by steam, or by waste heat supplemented by steam. When waste heat is used, the heated air is drawn from the kilns by fans and admitted to the tunnel at the delivery end; passing over the ware it gathers moisture and becomes more and more nearly saturated as it approaches the receiving end where it is exhausted by a second fan. In some driers of this type the temperature is permitted to vary with the temperature of the cooling kilns; in others, the temperature is controlled by the use of supplemental steam heat as well as by adjustment of fan speed and dampers and by admitting atmospheric air to the duct system. Supplemental furnaces are sometimes employed to supply additional heat at times of low production and when insufficient waste heat is available from the kilns. At least one of the manufacturers has installed automatic oil-fired furnaces for supplying heat to the tunnel drier in order to escape the temperature variations likely to occur when waste heat is used for this purpose.

60 In the steam driers the heat is supplied from radiating coils either distributed throughout the length of the tunnel or concentrated toward the dry end, and air is circulated by fans. The more modern driers of this type may be subdivided into graded heating zones with separate air circulation and control of both humidity and temperature.

61 Hand-made brick are almost invariably dried on drying floors; sometimes, however, only preliminary drying is done on the floors and the process is completed in a drier of the tunnel type. Machine-made brick are more commonly dried in tunnel driers, although with a number of manufacturers machine-made brick are dried exclusively on drying floors. Special shapes except the smaller sizes are also commonly dried on hot floors where they may be watched and frequently turned as required to avoid warpage; the drying is sometimes completed in the tunnel drier. Drying on floors usually requires three or four days for hand-made brick and about the same length of time for brick made by the machine-made wet process. Brick made by the dry-press processes, if dried at all before setting in the kilns, are allowed only

from 12 to 24 hours. Special shapes, if relatively thin, are sometimes dried more rapidly than the standard brick. The more massive shapes require much longer time, which for some shapes runs into several weeks. Generally only standard brick and the smaller shapes are dried in the tunnel driers, and the drying time varies with different manufacturers, with the character of clay and the shape, from 24 to 48 hours.

62 With some manufacturers the conditions and time of drying are fairly definitely scheduled for both hot floors and tunnel driers. With others, however, the drying is governed almost entirely by inspection of the ware, and this is invariably the case with the larger shapes. In the more modern tunnel driers, drying of the standard brick is generally conducted on a definite schedule and continuous records kept of temperature and humidity conditions. The temperatures in the tunnel driers will vary from slightly above atmospheric to about 140 deg. fahr. (60 deg. cent.) at the receiving end, and up to perhaps 250 deg. fahr. (121 deg. cent.) at the delivery end of the tunnel.

63 The dried brick from the hot floors are loaded on to cars or wheelbarrows for transfer to the kilns; the cars of brick from the driers are merely rolled into the kilns.

Burning

64 The kilns most commonly used in burning first-quality firebrick are of the circular down-draft and rectangular down-draft periodic types. The circular kilns are fired through doorways spaced around the periphery. In the rectangular kilns fireboxes are located either in the ends or at the sides of the kiln. In the more modern plants all kilns are of the open-bottom type in which permanent flues are constructed below the kiln floor with frequent downtakes to induce spreading of heated gas through the kiln.

65 In setting the kiln the brick are piled on the 2½ by 9-in. face, one above another, spaced slightly apart to provide gas circulation. At intervals of about four to ten courses one or two courses of brick are set at right angles to serve as ties, and the pile is carried to the equivalent of 25 to 30 courses in height, with the better-quality brick and special shapes set at the top. The piles are generally arranged in what are known as "benches" about 2 to 4 ft. wide extending across the kiln and spaced apart to provide gas circulation. Ducts extending from side to side through the ware are left opposite peepholes which are provided in the sides of the kiln for observation of burning conditions. Some of the manufacturers have made studies of heat distribution through their kilns, and in setting the brick give special attention to securing uniformity of heat distribution. The temperature variation from the top to the bottom of the usual down-draft type of

kiln is rarely less than 150 to 200 deg. fahr. (83 to 111 deg. cent.) the lowest temperature being nearest the bottom of the kiln and near the walls between firing doors.

66 In a number of plants, kiln temperatures obtained in burning first-quality brick are gaged entirely by eye and the completion of firing determined by the appearance of the ware and the shrinkage observed. Test brick are set conveniently near openings in the sides of the kiln so that they may be withdrawn for inspection toward the finish of the burn. Probably in the majority of plants, pyrometric cones are used for temperature determination. Generally these cones are set about 24 or 30 in. above the floor of the kiln. The test brick are set at the same level. Some manufacturers set cones in several places through the ware, at different points near the bottom and between the doors, and at about mid-height of the kiln. These cones, however, are generally exposed to the free flow of the gases and, therefore, subject to much earlier softening than would occur under the same temperature conditions if they were completely screened or boxed. When the softening of the cones indicates that the desired temperature has been obtained, test brick are withdrawn as a final check on the burn. In recent years, however, a growing number of manufacturers have come to use recording, and in some plants also optical, pyrometers as supplementing cones for following kiln temperatures. In general, the recording thermoelectric pyrometers do not register the actual temperature within the kiln but show relative temperatures and are used chiefly to afford a basis for temperature control. In some plants such pyrometers are used only through the water-smoking period; in others, they are installed as a permanent fixture in the kilns. The optical pyrometer affords the only definite measurement of kiln temperature. This instrument, however, can only be employed in the hands of skilled men. However, it is now coming into use by a number of progressive manufacturers as a check on other temperature indications.

67 The kiln burning is usually started with a relatively slow fire which is continued with gradually rising temperatures through the so-called water-smoking period which, with different manufacturers and varying with the character of clays and type of ware, will extend from 24 to 72 hours. In this period, during which the kiln temperature is raised to around 500 or 700 deg. fahr. (260 or 371 deg. cent.) and with some manufacturers to 900 deg. fahr. (482 deg. cent.), the remainder of the mechanically held moisture is driven off and with it some of the sulphur, also possibly some of the chemically combined moisture and gases from such organic matter as has been contained in the clay. The rate of fire is then increased and the kiln temperature brought to the maximum in from thirty-six hours to four or five days, varying with the manufacturer and the ware being burned. After attaining the maximum temperature the fires are generally continued and this

temperature held for from twelve to forty-eight hours, after which the kiln is completely closed and permitted to soak for another twelve and sometimes for forty-eight hours, after which vents are opened in the crown of the kiln and gradual cooling continued for about twelve to twenty-four hours, when the kiln is completely opened and in the next day or two is ready for unloading. Fig. 7 shows typical thermocouple records for the periodic down-draft kiln. The rate of temperature change and the maximum temperature attained will, of course, vary with different manufacturers. In general, however, the maximum temperature applying to any large percentage of the brick in the ordinary first-quality fireclay brick kiln will not exceed 2300 deg. fahr. (1260 deg. cent.). Many of these brick are burned at still lower temperatures.

68 One of the larger manufacturers is using the continuous-chamber type of kiln; Curve 3 of Fig. 7 is taken from the records

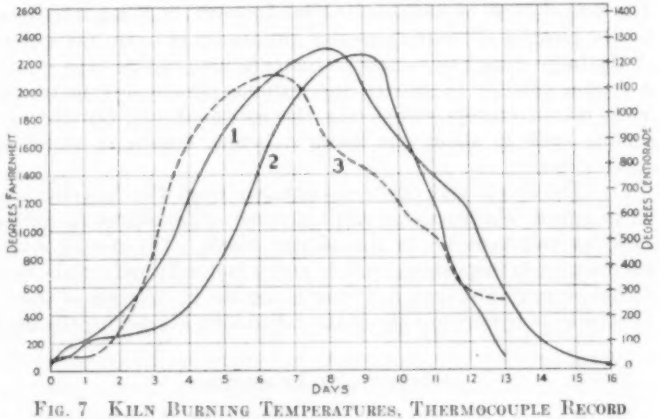

FIG. 7 KILN BURNING TEMPERATURES, THERMOCOUPLE RECORD

for one of these kilns. The pyrometer thermocouples from which the records are taken are located in the crown of the kiln but as the couples are protected by refractory tubing or tile the temperatures recorded are somewhat below the actual maximum at the top of the kiln. The temperatures at the bottom of the kiln are probably 50 to 100 deg. fahr. (28 to 56 deg. cent.) lower. In this instance the waste heat from the kilns under fire supplies the heat for the water smoking. At the end of this period the direct fires are lighted in the kiln and the temperature is raised rapidly to about 1800 to 1900 deg. fahr. (982 to 1038 deg. cent.), after which it is raised more slowly to the maximum around 2200 deg. fahr. (1204 deg. cent.) or slightly higher. In the case of the regular periodic kiln the thermocouple is protected in the same manner and its records are subject to the same sort of correction. The other curves are for kilns of regular down-draft type, No. 2 for the ordinary burn, No. 1 for a specially high burn.

69 In recent years there has been much development work in adapting the continuous tunnel type of kiln to burning first-quality brick. Three or four manufacturers are now using tunnel kilns, and others have installations under way. This type of kiln consists essentially of a tunnel which may be from 100 to 400 feet in length through which small cars with refractory floors carrying the molded and partly dried brick are pushed continuously. There are generally three zones in the tunnel; a preheating zone for final drying, water smoking, and oxidation; a firing zone; and a cooling zone. Waste gases from the firing zone serve for preheating in the first zone. Atmospheric air is admitted for cooling in the final zone.

70 Apparently there is a demand for brick of wide variation in burn, so that manufacturers using the periodic type of kiln claim in general to suffer no inconvenience in disposal of their product on account of variation in the different parts of the kiln. The tunnel type of continuous kiln, however, is being introduced for the purpose of securing greater uniformity of burn. On account of the more complete control of temperatures in this type of kiln, it is also possible to shorten the burning period very materially, some manufacturers completing the entire drying, burning, and cooling cycle in about one hundred hours; others, in materially shorter time.

71 The brick and tile are unloaded from the kiln by hand on to spring wheelbarrows and conveyed either direct to railway cars for shipment or to the storage sheds. The use of wheelbarrows for transporting brick is now practically universal. Some manufacturers, however, are considering the installation of electric lift trucks to eliminate some of the rehandling in stacking the brick.

Grading and Inspection

72 Some of the manufacturers dispose of their brick as of a single burn, considering that temperatures throughout their kilns have been adjusted to sufficient uniformity for this requirement. Others supply as many as four or five different burns ranging from hard to light, the hard burn representing material burned at a temperature of about 2300 to 2400 deg. fahr. (1260 to 1316 deg. cent.) and the soft burn at a temperature of 2200 deg. fahr. (1204 deg. cent.) or sometimes lower. The grading is generally determined by the position of the brick in the kiln. The hardness is by some manufacturers determined by the "ring" of the brick. It cannot be judged by color alone as that is markedly affected by the atmosphere of the kiln as well as by the contact of gases and flame.

73 Some manufacturers confine their periodic tests to the raw materials and unfired brick, and make tests only occasionally on the final product. Some of these send the unburned brick weekly to the Mellon Institute for test of fusion and analysis, and

reports of these tests are available before the brick are through the kiln. With these manufacturers the examination of the burned brick is confined to inspection for physical defects and other points of unfavorable appearance such as that of unusual iron content. Some manufacturers make a practice of checking their product by fusion and analysis on both raw and burned materials monthly. Others make various tests, fusion, spalling, load, and reheat, monthly, and others make such tests quarterly. Some manufacturers confine their regular tests exclusively to the finished product, and make fusion tests and analysis on one or two brick from every kiln. With some of these manufacturers the results of the fusion tests are available within a few hours or on the following day. With others there is a lapse of ten days to two weeks before test results are known, dependent, of course, upon the character of test employed.

74 Within the past five years an increasing number of firebrick manufacturers have been devoting attention to the sizing of their brick. The majority of manufacturers are now prepared to supply brick of size within the standard range accepted by the Refractories Manufacturers Association. As stated in describing the detail of the processes, within the past three years two of the manufacturers have brought out special sizing machines. These manufacturers guarantee variations less than one-half this range. Many other manufacturers by devoting more careful attention to sorting their clays and by more accurate control of their processes, especially as affecting the sizing and mixing of their clays and spacing of wires on wire cutting machines, have also made material improvement. Uniformity is more readily secured in the machine-made brick. In the hand-made brick the product is so largely dependent upon the human element that real improvement can only be had through exercise of greater care and intelligence on the part of the individual.

Developments in Firebrick Manufacture

75 Although firebrick manufacture is one of our older industries, generally considered to have been founded in New Jersey [25] about 1812, it has not had the degree of development nor, until within a very few years, the technical attention, from which many of our industries have benefited. Recently, however, there has been a more general recognition on the part of the manufacturers of the importance of research and, in the last two or three years very marked advances have been made in the introduction of more scientific methods and in the use of systematic control in manufacturing processes. In this connection a number of manufacturers report great benefit derived from the work of the Mellon Institute.

76 The firebrick industry is today facing about the same stage in its development as the spark-plug porcelain industry some ten years ago. In the case of the porcelain manufacturers, the problem was met first by intensive study to determine the actual requirements of a spark plug and second by the searching out and development of materials to meet these requirements, until the modern spark-plug porcelain presents a product which is almost perfect for its purposes. Firebrick manufacturers can well follow a corresponding course, and the promise of attainment is equally sure.

77 Up to a few years ago, at least as affecting the development of boiler-furnace refractories, there was an almost total absence of coöperation between manufacturer and user. Such coöperation is a first essential to a clear understanding of the requirements of refractories. The most substantial progress in this respect will be made by continuance of the coöperative endeavor which is now making its beginning.

78 It would seem probable that in the firebrick industry, while to search out and develop better materials for special refractory requirements is of very great importance, it is of even greater importance to industry at large that the best use be made of the materials at present known and commercially available.

79 One of the chief problems of the firebrick manufacturer has seemed to be the maintenance of the standard of quality set by his best production. The sources of this lack of uniformity are found at every stage of his manufacturing process. The first variable is encountered in the sorting of clays at the mine. Not only must concentrations of impurities be avoided but it is of almost equal importance that there be no mingling of clays which though deposited one immediately above the other in the seam are of radically different characteristics. Hand sorting is still the best available method and this must frequently be done in dimly lighted mines.

80 One of the western manufacturers has been able to effect marked improvement in iron content of clay by magnetic separation.

81 In the processes having to do with proportioning of raw materials at the plant, there is widespread need for more careful and intelligent sorting, storing, and measuring of these materials. Also, more detailed study can well be devoted to effects of sizing. It would appear entirely logical that the sizing of materials in brick manufacture would be almost as important as in concrete and that similar principles should apply to the screen analysis. The experience of some of the firebrick manufacturers would seem to bear this out.

82 Segregation of sizes and of materials is a very serious problem in bin storage. A number of manufacturers report diffi-

culty from that cause and at least one manufacturer has completely abandoned bin storage to escape this segregation.

83 Both structural weakness and poor refractory qualities are likely to result if the mixing and kneading processes are not thoroughly done. Brick are commonly mixtures of clays. To secure the maximum stability of bond the materials must be uniformly distributed and intimately mixed.

84 Hand molding necessarily involves variations of size and texture and, although with the skilled workmen these variations under favorable conditions may be kept within reasonable limits, constant uniformity can hardly be expected except in a completely machine-made product, made up very largely of previously calcined materials.

85 The rate at which the molded brick can be satisfactorily dried and carried through the earlier stages of burning in the kiln is governed by the nature and texture of materials of which it is composed. For brick made largely of materials previously calcined, these processes may apparently be conducted fairly rapidly, but for the average clay containing a normal percentage of impurities and especially when the molding is done by one of the wet processes, careful scheduling and control are essential to avoid injury to the brick from strains resulting from too rapid shrinkage and too rapid evolution of moisture and gas. To avoid excessive strains and surface cracking, the drying must take place from the inside out and there should be such control of temperature and humidity as will bring this about.

86 Much improvement can be had from more scientific control of burn, not only through these earlier stages but also at the period of highest firing. Very few manufacturers have definite knowledge of the variations of temperature in their kilns or even of the maximum temperatures attained. In many instances no temperature measurements are made. Manufacturers who use cones rarely appreciate that in setting the cones in open channels they are subject to early softening from exposure to flame and gases or that cones are standardized for fusion temperatures attained within an hour and that, at the rate of heating found in the commercial kiln, these cones soften at 100 to 200 deg. fahr. (56 to 111 deg. cent.) below their nominal temperature. "Cones" are measures of heat effects," and fortunately for many purposes their indications are a sufficient guide to the burn of the brick. However, there are changes which under ordinary conditions of atmosphere and pressure can only be brought about by application of definite temperature and accordingly there are for every clay, definite temperatures at which the clay should be burned for the best development of its refractory qualities as applying to specific classes of service. Considerable improvement would result if brick

were burned only at predetermined temperatures and then definitely classified according to the temperature of burn.

87 It would result in greater semblance of uniformity and assist very materially toward coöperation between manufacturer and user if brick branding were standardized to indicate the clay used and process of manufacture. Some manufacturers have used identical branding for brick made by several different processes. The confusion resulting from such practice can readily be appreciated.

SERVICE CONDITIONS AND PROPERTIES OF REFRACTORIES

88 In the early installations of stationary boilers, in fact prior to about 1905, furnace temperatures were so low and general service conditions so mild, as compared with those in the steel industry for example, that it was commonly considered that any firebrick was good enough for a boiler setting. Anthracite buckwheat was the only fuel on which forced draft was used to any extent. Little attention was given to limitation of air supply and at the higher rates of operation, which rarely exceeded 150 per cent of the nominal boiler capacity, combustion largely occurred within the immediate zone of the heating surface. Boilers were set about as near the floor as their design permitted; for boilers of the horizontal water-tube type the front headers were 8 ft. above the floor where bituminous coal was to be burned and 5 ft. above the floor for anthracite. These settings gave actual furnace heights of only 3 to 6 ft. The maximum boiler size was about 600 hp., for which the furnace was about 12 ft. in width by 8 or 9 ft. in horizontal depth.

89 The forced-draft underfeed stoker for bituminous coal was introduced in 1904. This was followed by the forced-draft traveling grate for anthracite in 1912 and by the commercial adaptation of the traveling grate to the burning of midwestern bituminous coals under forced draft in 1920. In every case these stoker developments have brought increased rates of combustion, with better control of air supply, thus tending to higher furnace temperatures.

90 With the growing appreciation of the desirability of completing combustion before the heated gases reach the boiler surface, there has also been continual increase of furnace volumes with corresponding increase in wall area directly exposed to the burning fuel and products of combustion. This movement, beginning about 1907 when the maximum uptake header height above the floor for the horizontal water-tube boiler was increased to about 9 ft. 6 in., received special impetus with the commercial introduction of pulverized coal as a steam-boiler fuel about 1918. At that time setting heights of 12 to 13 ft. under the headers for boilers of the horizontal water-tube type had become quite general.

Within the next two or three years, however, setting heights jumped to 18 and 20 ft. and more recently to 40 ft. The resulting increase in ratio of radiating wall area to area of exposed heat-absorbing surface brought still further increase of furnace temperature. There has, at the same time, been a marked increase in boiler size, and furnace dimensions approximating 24 by 30 ft. in horizontal section and 25 to 30 ft. in height are now not at all uncommon.

91 The improvements in capacity and efficiency of coal-burning equipment have been paralleled in the oil-burning field by the adaptation, in 1919, of the mechanically atomizing forced-draft type of burner to the requirements of the stationary boiler. It will be appreciated that these developments in boiler, combustion equipment, and furnace have necessarily made increasing demands upon the refractories used for boiler-furnace construction.

Refractory Problems

92 The earliest problem of serious moment was probably that of clinker adhesion on the side walls and bridgewalls of furnaces burning bituminous coal. The accumulation of the clinker interfered with combustion, and its removal, generally by slice bar or chisel, was accompanied by fracture and abrasion of the brick. Difficulties of a similar nature, but much less severe, were experienced in burning anthracite. With the introduction of the forced-draft traveling grate, however, the clinker problem became serious with anthracite also. Various schemes of prevention were tried: the admission of low-pressure steam and air beside the wall below the clinker line; the use of water-cooled cast-iron wall boxes, water-cooled steel boxes or tubes in circuit with the boiler circulation, and silicon-carbide brick inset in the wall at the clinker line; the admission of air blown through the brickwork into the furnace, both along the clinker zone and somewhat above it. By about 1920-1922 measures for avoiding clinker adhesion had been tentatively standardized approximately as follows: For underfeed-stoker furnaces burning bituminous coal, cast-iron wall boxes ventilated by air blown into the combustion zone, supplemented by ventilated brickwork also discharging the cooling air into the furnace; for forced-draft traveling-grate furnaces burning bituminous coal, steel boxes or tubes cooled by boiler circulation set in the walls at the clinker line; for forced-draft traveling-grate furnaces burning anthracite, silicon-carbide brick set in the side walls in the clinker zone immediately above the stoker ledge plate.

93 The 1912 report [15] of the Prime Movers Committee of the National Electric Light Association suggests the possibility of melting the most refractory fireclays at the temperatures to be had in furnaces operated with excess air limited to give as high as 15 per cent CO_2 in the products of combustion. The 1914 report [16] of this committee, however, recognizes that other quali-

ties, resistance to temperature change and resistance to the fluxing and eroding action of slags and the abrasion of clinker, are more important than mere refractoriness. The higher rates of combustion and the more efficient control of combustion were giving furnace temperatures at which, with certain bituminous coals, the clinker and fine ash deposited on the brickwork ceased merely to adhere but melted to a very fluid slag, which penetrated and dissolved the structure of the brick and washed away deep grooves in flowing down the walls. The slag problem became increasingly serious with the introduction of pulverized coal for, with that mode of combustion the entire fuel feed is in suspension in the furnace instead of merely the relatively small percentage that is blown off the grate with stoker firing. In pulverized-coal furnaces, the slag problem has been met, first, by admitting air through the walls at points adjacent to the areas of greatest slag accumulation and, later, by the installation of so-called "slag screens," followed by an increasing use of water-cooled and steam-cooled surfaces in the furnace walls. The same general methods of wall cooling are being adapted to the requirements of stoker-fired furnaces of both the underfeed and the traveling-grate types.

94 In the earlier installations, spalling was sometimes a factor in brickwork deterioration, especially where subject to repeated sudden cooling, as around furnace doors. Spalling became important only with the introduction of the suspended arch in the chain-grate furnace. With increasing temperatures it has become a serious factor in the deterioration of modern furnaces of all types. In a number of oil-burning furnaces, in the parts of the walls most subject to spalling, the brick are spaced slightly apart to admit air between them, and thus by cooling and oxidation of the immediately adjacent atmosphere to reduce the severity of conditions to which the brick are exposed.

95 With increasing furnace dimensions, as well as increasing temperatures, failures from the mechanical stresses of expansion have also become more in evidence. This class of failure, however, cannot be charged to the quality of the refractory; it can only be relieved by better design.

Refractory Surveys

96 In 1922, through the coöperation of a number of power producers and the United States Bureau of Standards, a survey was made of firebrick service in steam-boiler plants considered representative of the best operating practice throughout the country. An outline of this survey is given in Technologic Papers" of the Bureau of Standards, No. 279. The survey dealt primarily with coal-burning plants, although some oil-burning installations were included.

97 Reports were obtained from the plant operators in answer to a set form of questions covering the identification of refractories used in different parts of the furnace, the operators' opinion as to the relative influence of different factors — spalling, fusion, clinker removal, slag erosion — in causing failure, the type of installation, and such conditions of operation as grade of fuel used, boiler load factor, and rating. On account of the great diversity of conditions to be considered it proved impossible in so brief a survey to determine very definite relationships. There were too many undetermined or only partly determined variables. The apparent order of importance of the different factors considered in causing failure in boiler-furnace side walls is given in Table 2. The percentages shown are averages rated in proportion to the installed boiler capacities of the individual returns. The great majority of boilers covered by this survey were of about 600 hp. rated capacity. Perhaps the most important results of the survey lay in the extent to which it emphasized the need for more systematic, thoroughgoing attention to firebrick problems.

TABLE 2 CAUSES OF SIDE-WALL FAILURE, 1922 AND 1924 SURVEYS

Kind of Fuel	1922	1924
Eastern bituminous (underfeed stokers)	45% Slag adhesion 21% Spalling 18% Erosion 16% Fusion	1 Slag adhesion 2 Failure of structure 3 Erosion 4 Spalling
Midwestern bituminous (traveling grates)	90% Erosion 10% Slag adhesion	1 Erosion 2 Spalling 3 Failure of structure
Anthracite	96% Erosion 2% Slag adhesion 1% Spalling 1% Fusion	1 Failure of structure 2 Spalling 3 Slag adhesion
Oil	95% Spalling 5% Fusion	1 Spalling 2 Erosion 3 Failure of structure

98 In 1924, at the suggestion of C. F. Hirshfeld of The Detroit Edison Company, a second survey [32] was undertaken by the Bureau of Mines. This survey is being conducted by personal inspection of representative plants. R. A. Sherman, the technical representative of the Bureau on this field work, has rendered a preliminary report of his findings in thirty-four different plants visited. An abstract of this report was given to the technical press in August of this year.

99 The relative importance of different factors in causing failure of firebrick in boiler-furnace side walls as observed on the 1924 survey is presented in Table 2 to afford comparison with the corresponding findings of 1922. The average size of boiler was much larger and the effects of mechanical stress, appearing in the table as "Failure of Structure," assumed greater proportions than in the 1922 survey. Although, as there was no special classification for mechanical failures in the 1922 survey, undoubtedly many such failures were at that time classified as spalling.

100 Both surveys report slag adhesion as the principal cause of failure in boiler furnaces burning eastern bituminous coal, the

actual destruction of the brick being in this case due to the removal of the slag accumulations. Deterioration of this type is, of course, increased by spalling tendency in the brick. "Fusion" reported in the 1922 survey is undoubtedly the fluxing action of slags and this cause of failure might more appropriately be combined with "Erosion."

101 For midwestern bituminous coal erosion appears as the dominant cause of failure. In 1922 erosion was also reported as the most important cause of side-wall failure in anthracite-burning furnaces. In 1924 however, failure of structure due to expansion stresses was presented as of somewhat greater importance in the installations covered at that time. For the mechanically atomizing oil-burning furnace spalling was reported as the most important form of deterioration in both 1922 and 1924.

102 While the number and diversity of factors influencing the life of the brickwork have also in the case of the 1924 survey prevented definite evaluation of these factors, the greater detail and uniformity of viewpoint in the 1924 survey have provided better bases of comparison and afforded a surer indication of trends.

103 The 1924 survey points rather strongly to improved life of refractories per unit of output, for any one type of stoking equipment, accompanying an increase in rate of output within the range of average loads considered, 100 to 200 per cent of the rated capacity of the boiler. This, of course, is merely a generalization and is known to be inapplicable to some conditions of extremely fusible ash. Higher boiler load factors also show strong indications of bettering firebrick life. A factor, the effect of which it seems almost impossible to obscure, is that of ratio of hours banked to hours of actual operation. Comparisons of brickwork maintenance requirements for the range of banking to operating ratios observed on the 1924 survey for underfeed-stoker installations burning eastern bituminous coal and for natural-draft chain-grate installations burning midwestern bituminous coal point to increase of maintenance per unit of output accompanying an increase of this ratio up to about 0.8 or 1.0; in other words, accompanying the reduction in operating hours from 22 or 23 to 12 or 13 per day. With further reduction in daily operation, however, there is apparently a checking in rate of deterioration and, with midwestern coals, there may even result an actual reduction in rate of brickwork maintenance per unit of output. While undoubtedly there are many unrecorded factors influencing the life of brickwork in these instances, it is believed that the trend shown is more than accidental. It would be expected that the longer the banking period the cooler the brickwork would become before the fire was again broken out and therefore the wider the range of temperature and the greater the thermal stress to which the brickwork would be subjected in the daily operating cycle. On the other hand,

banking the boiler for approximately half the day or longer probably affords sufficient cooling of the brickwork to effect very material reduction in slag penetration and erosion. If these apparent relations are approximately representative, then spalling has a more important place in the deterioration of stoker-fired boiler furnaces than has been generally supposed.

General Properties of Refractories

104 The properties of refractories are most conveniently interpreted in terms of their service characteristics. Those qualities which are most apparent in their direct effect upon service life are resistance to fusion, load-carrying capacity at furnace temperatures, permanence of volume at furnace temperatures, mechanical strength, resistance to clinker adhesion and removal, resistance to fluxing and erosive action of slag and gases, resistance to spalling, and thermal expansion. Some of these qualities are more or less interdependent. There are other fundamental though possibly less obvious qualities or properties upon which many of these are in some measure dependent: thermal conductivity, specific heat, and diffusivity which is a function of these two, elasticity, absolute density, porosity, permanence of microstructure at furnace temperature, and chemical activity at furnace temperature.

105 All of these properties are related not only to temperature and heat but to atmosphere. The atmosphere in contact with the heated brick may effect either oxidation or reduction of its impurities and, so, markedly influence all of its characteristics.

Furnace-Wall Temperatures

106 Furnace-wall temperatures are in general more affected by radiation than by heat convection from gas currents and, where not shielded by the opacity of the gas, may frequently exceed those of the gases immediately in contact by 100 to 200 deg. fahr. (56 to 111 deg. cent.) or more. In underfeed-stoker furnaces using air at atmospheric temperatures for combustion, while the fuel-bed temperature may in spots reach 2900 deg. fahr. (1593 deg. cent.) and sometimes higher, on account of the cross radiation to cooler masses of fuel and opacity of intervening gases, the wall-surface temperatures will rarely exceed 2300 to 2500 deg. fahr. (1260 to 1371 deg. cent.). Air admission along the slag line may keep the wall temperatures still lower. When the air for combustion is preheated, wall temperatures may be somewhat higher. In the forced-draft traveling-grate furnace burning midwestern bituminous coal, fuel-bed temperatures will frequently be more uniformly high and refractory surface temperatures reach 2700 to 2800 deg. fahr. (1482 to 1538 deg. cent.) in walls and arches. In the same general type of furnace equipped with front and rear

arches burning anthracite, refractory-surface temperatures between arches may reach as high as 2700 deg. fahr. (1482 deg. cent.). In the pulverized-coal furnace and in the mechanically atomized oil-burning furnace the highest temperatures follow the zone of maximum intensity of combustion, which in turn is governed by the mixing ability of the burner and the rate at which it is operated. Flame temperatures reach a maximum of about 2800 deg. fahr. (1538 deg. cent.) in the usual types of pulverized-coal furnace and about 2900 deg. fahr. (1593 deg. cent.) in the mechanically atomized oil-burning furnace. These temperatures are of course affected by the extent and arrangement of exposed water- and steam-cooled surface but the effect of such surface upon the temperature of the zone of most intense combustion may be not

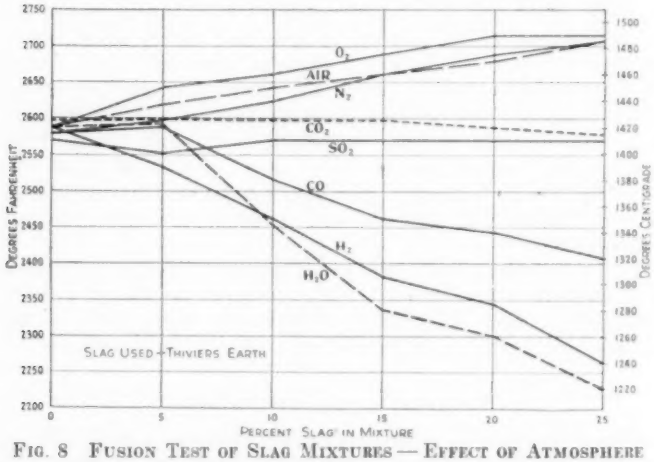

FIG. 8 FUSION TEST OF SLAG MIXTURES — EFFECT OF ATMOSPHERE

nearly as great as upon the mean furnace temperature. Wall surfaces may approach within 100 to 200 deg. fahr. (56 to 111 deg. cent.) of the maximum temperatures even though separated from the flame by 3 or 4 ft. of cooler gases. In the oil-burning furnace, wall-surface temperatures are often the result of direct flame impingement and may approach within 100 deg. fahr. (56 deg. cent.) of the temperature of the flame.

FURNACE ATMOSPHERE

107 Fieldner, Hall, and Field [39] in studying the determination of fusion temperature for coal ash, which fundamentally is of the same general character as fireclay but with excessive impurities, found the atmosphere in which the ash was heated by far the most important factor in influencing the observed result. Variations in softening temperatures for the same ash, due to differences in atmosphere, ranged as high as 713 deg. fahr. (396 deg. cent.).

These investigators found that for the particular grade of ash tested the highest softening points were obtained either in an atmosphere of air or in a very strongly reducing atmosphere in which the iron oxides were completely reduced to metallic iron before softening of the ash began. The lowest softening temperatures were obtained in atmospheres of mixed gases in which reduction of ferric oxide proceeded mainly to ferrous oxide. A. V. Bleininger in 1915 in examining the ash of a Kansas coal found a fusion point of 2358 deg. fahr. (1292 deg. cent.) in air and of 1927 deg. fahr. (1053 deg. cent.) in a reducing atmosphere of equal parts of hydrogen and water vapor. Vickers and Theobald [20] in England have conducted similar investigations on materials of somewhat lower impurity content and approaching more nearly the composition of normal clays. Iron in the form of Thivier's earth was mixed in varying amount with fireclay, and softening temperatures determined in atmospheres of nitrogen, air, oxygen, hydrogen, carbon monoxide, carbon and sulphur dioxides, and

TABLE 3 ANALYSIS OF CLAY AND THIVIER'S EARTH
(Vickers and Theobald)

		Fireclay	Thivier's earth
Silica	SiO_2	72.36	83.18
Titania	TiO_2	0.04	....
Alumina	Al_2O_3	17.04	2.30
Ferric oxide	Fe_2O_3	0.35	9.87
Ferrous oxide	FeO	....	0.33
Manganese oxide	MnO	....	0.04
Lime	CaO	1.32	0.83
Magnesia	MgO	0.15	0.12
Potash	K_2O	4.81	0.58
Soda	Na_2O	1.11	0.22
Ignition	....	2.62	2.82

water vapor. The effects of these various gases upon the softening temperature are shown in Fig. 8. For the particular materials investigated, the analyses of which are given in Table 3, water vapor has the most pronounced effect, reducing the softening temperature for the mixture of highest iron content 479 deg. fahr. (265 deg. cent.) below that obtained with this mixture in an atmosphere of either air or nitrogen and nearly 360 deg. fahr. (200 deg. cent.) below that obtained in an atmosphere of CO_2. The atmosphere of carbon monoxide is also shown as lowering the softening point very markedly. As the service characteristics of firebrick are largely governed by the structure, it will be appreciated that any condition which tends to lower the softening point and so affect the structural stability will largely influence all of its properties.

Fusion

108 Fusion temperatures for a number of fireclays from different parts of the country are given in Table 1. These temperatures are all higher than the temperatures to which the molded brick are subjected in the usual steam-boiler furnace. In fact, first-quality refractories have rarely failed from actual fusion unaided by the

fluxing action of constituents in the fuel burned. This type of failure has been practically limited to the oil-fired furnace, and even with oil firing it is believed that in many instances failures which have appeared to be due solely to fusion of the refractories have been materially assisted by the mineral constituents deposited on the wall by the oil. Analyses of fuel oils have shown the presence of iron and other minerals, and in some instances alkalis in excess of 0.02 per cent. The analysis of ash from petroleum coke given in Table 4 is representative of Mid-Continent crude.[21] The mineral constituents are, of course, subject to variation. A more highly corrosive material, as affecting the ordinary fireclay brick, would be difficult to suggest. An analysis of slag secured in connection with the 1924 survey, from the bottom of an oil-fired furnace using mechanically atomizing burners, is also shown. The iron and alkali contents of the slag are materially higher than could possibly have been derived from fusion of the refractories alone. The total ash content of fuel oil is very small, expressed as

TABLE 4 ANALYSIS OF FUEL-OIL SLAG

	Mid-Continent petroleum-coke ash	Panuco oil-fired-furnace slag
Silica	43.99	49.00
Alumina	1.26	25.90
Iron oxide	27.71	8.00
Titania	....	0.97
Lime	22.45	0.90
Magnesia	....	1.20
Alkalis	....	4.20
Sulphur trioxide	1.15	0.0
Vanadium oxide	....	6.1
Sodium chloride	3.32	....

a percentage, two or three-hundredths of one per cent; but, as a single burner frequently discharges 1000 lb. of oil per hour, the actual quantity of ash may be quite large.

Load-Carrying Capacity

109 At atmospheric temperatures, ordinary first-quality brick show crushing strengths ranging from about 1200 to 4500 lb. per sq. in. with moduli of rupture under transverse load of 300 to 1500 lb. per sq. in. — the dense, finely ground, hard-burned products giving the maximum strength, the softer-burned brick containing a high percentage of flint clay showing the lowest values.

110 The load-carrying capacity of firebrick at furnace temperatures, while limited by the quality of materials from which the brick are made, is largely governed by the initial density of the brick structure and the temperature at which it has been fired in the kiln. In laboratory tests, brick are usually subjected to a load of 25 lb. per sq. in. and the temperature raised according to predetermined time schedule. Other loads, 40, 50, and sometimes below 25 lb. per sq. in., have been used by various investigators for special determinations. In Fig. 9 the specimen at the right

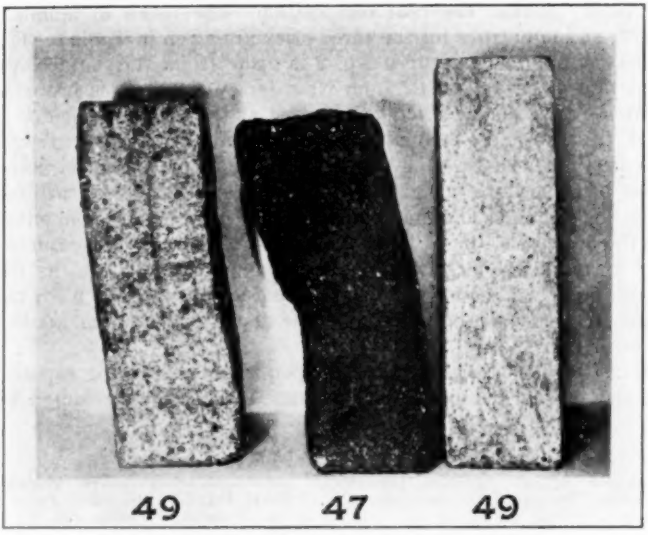

FIG. 9 LOAD-TEST SPECIMENS, FIRECLAY BRICK

is a standard 9-in. brick; the other two specimens are characteristic of deformation under 25 lb. load.

111 Fig. 10 shows typical characteristics[22] of firebrick of the normal range of commercial burn heated under a load of 25 lb. per sq. in. This figure further illustrates by the curves marked No. 2 and No. 82 some of the diversity in characteristics of clays

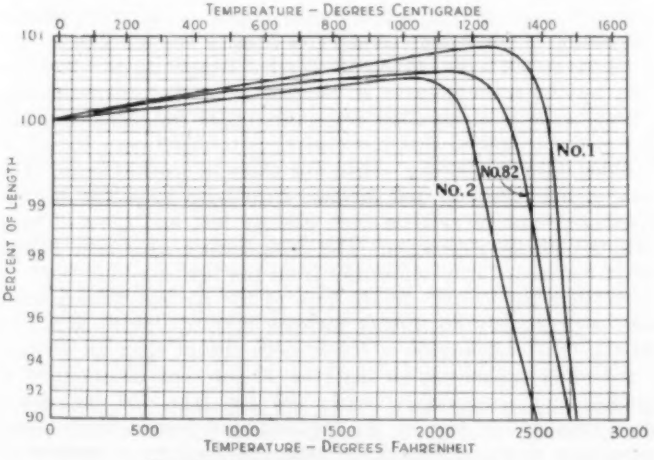

FIG. 10 LOAD TEST, 25 LB. PER SQ. IN.

of closely similar chemical composition. The ratios of alumina, silica, and impurities for the three brick are given in Table 5. The brick represented by curve No. 2 is made from Maryland clays, and that of curve No. 82 from New Jersey clays. It is probable that the New Jersey brick was burned at the higher temperature and this, in part, accounts for the difference in load-carrying capacity. Curve No. 1 is for brick made from California clays. This brick sustains the load at much higher temperature without deformation and shows much more abrupt compression than either of the other two brick. This characteristic is also to be accounted for as due very largely to higher temperature of burn, as the California brick shows a higher alumina-silica ratio which for the same temperature of burn would tend to give a more gradual deformation.

112 Fig. 11 shows the effect [22] upon the load-carrying capacity of varying the temperature of burn. The brick represented by

TABLE 5 FIRECLAY BRICK

Brick, Number	Silica, Per cent	Alumina, Per cent	Impurities, Per cent	Fusion			Porosity, Per cent
				Cone	Deg. fahr.	Deg. cent.	
1	50	45	5	34	3164	1740	..
2	58	38	4	33	3128	1720	..
4	60	38	2	32	3101	1705	..
5	52	43	5	32	3101	1705	22
7	55	41	4	33	3128	1720	17
10	54	41	5	33	3128	1720	28
12	20	73	7	38	3308	1820	34
20	46	50	4	36	3236	1780	30
23	55	41	4	33	3128	1720	14
28	53	43	6	33	3128	1720	23
29	53	43	6	33	3128	1720	..
33	51	45	4	33	3128	1720	23
58	52	43	5	32	3101	1705	13
82	57	39	4	35	3200	1760	..
83	53	42	5	32	3101	1705	22

these curves are of a standard Pennsylvania brand, the essential composition of which is given in Table 5, brick No. 83. The several specimens tested were burned at different temperatures which are given by the manufacturer as ranging from that corresponding to Seger cone 8, about 2354 deg. fahr. (1290 deg. cent.) for the light medium burn to that corresponding to Seger cone 13, about 2534 deg. fahr. (1390 deg. cent.) for those of hardest burn. The brick of light-medium and medium-hard burn show the highest expansion, those of medium burn slightly less, and those of hard burn the lowest expansion. The temperature at which failure occurs increases fairly uniformly from the light-medium burn to the medium-hard burn. In the case of the hardest-burned brick there is practically no further increase in temperature of failure and the final compression is much more abrupt than in the case of the other specimens, so that the 10-per cent deformation is obtained at a temperature nearly 100 deg. fahr. (56 deg. cent.) lower than in the case of the medium-hard-burned brick. Corresponding characteristics have been observed by a number of

investigators.[23] It would seem probable that the decreased expansion of the brick of hardest burn is due both to the further development of mullite and to the greater conversion of uncombined silica to the cristobalite and tridymite forms and to impure quartz glass, and that the more abrupt failure of this brick is due to the crystallization of the mullite tending to a concentration of impurities in the uncombined silica which would be highest in the bond clay.* The more extended crystallization would give higher structural strength and increased refractoriness, but, with the concentration of impurities in the uncombined silica, apparently a eutectic is formed which at the final temperature attained in the load test is of very low viscosity.

113 Under the loads imposed by normal boiler-furnace construction, the average first-quality brick will show no appreciable

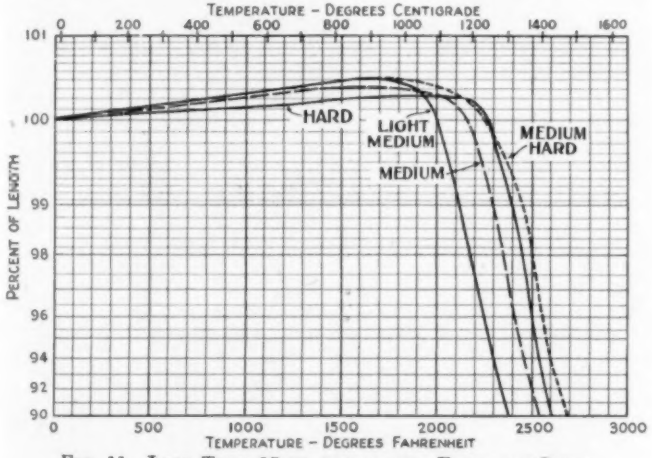

FIG. 11 LOAD TEST, 25 LB. PER SQ. IN., EFFECT OF BURN

deformation at temperatures below about 1800 deg. fahr. (982 deg. cent.). F. H. Norton,[24] in studying torsion stresses on refractory brick at 932 deg. fahr. (500 deg. cent.) using loads of 145 to 450 lb. per sq. in., found measurable elastic deformation at this temperature but no permanent set except just before failure. M. C. Booze and S. M. Phelps [25] conducted a similar investigation and found that for the particular brick tested, some of which had previously been fired at cone 12, the actual firing temperature being probably somewhat lower than 2498 deg. fahr. (1370 deg. cent.), plastic, permanent deformation would take place under stresses from 75 to 175 lb. per sq. in. at a temperature as low as 1175

* See Par. 14 and Fig. 3. It is, of course, only the tridymite and quartz glass which have lower coefficients of expansion than the original quartz but it is believed impossible, under the conditions described, to avoid formation of cristobalite.

deg. fahr. (635 deg. cent.). These stresses, however, are considerably beyond anything permissible over any very large section of boiler-furnace wall.

114 In a 23-in. solid brick wall without insulation, for a furnace operated under slight suction draft, temperatures above 1800 deg. fahr. (982 deg. cent.) might be expected to affect the brick to a depth of 6 or 7 in. from the fire face. In a properly bonded wall the load would be taken by the cooler brick and no serious deformation occur. However, if insulation were applied to a wall of this thickness or if the wall were 27 in. or greater in thickness, it would

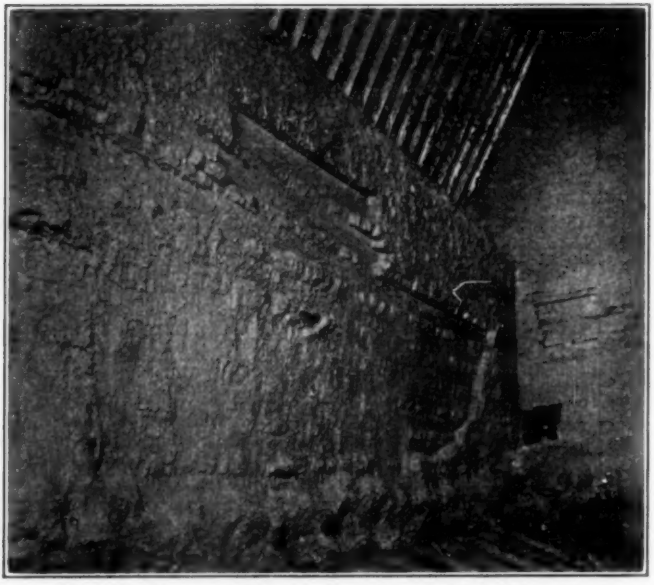

FIG. 12 FRONT-WALL EROSION, STOKER-FIRED FURNACE

be necessary to select one of the more highly refractory brick to avoid risk of failure if fire-face temperatures in excess of 2500 deg. fahr. (1371 deg. cent.) are to affect any very large area of the wall.

115 In actual boiler service, softening or deformation under load is much more common than fusion, although, except in sprung-arch construction, such deformation is rarely the sole cause of failure. Softening is generally a secondary result of shrinkage, spalling, or a combination of spalling and slag erosion. Fig. 12 gives an illustration of the latter type of failure in a stoker-fired furnace. It will be seen that the furnace lining, following the best practice of only a few years ago, was built up of three stretcher courses alternating with a single header course. The combined

action of slag and flame erosion and spalling apparently cut away some of the lower courses, which left a header course not only supporting the three stretcher courses above but with half the lower face exposed to the full heat of the flame. The brick used in this instance would probably begin to soften below 1900 deg. fahr (1038 deg. cent.) and, as the exposed ends would be heated at least 200 or 300 deg. fahr. (111 or 167 deg. cent.) above this temperature, the failure of the lower courses was naturally followed by progressive collapse of the entire lining. In sprung arches, failures due to deformation of the refractories, while occasionally accelerated by spalling and by slag erosion or fluxing, are generally attributable directly to the heating of the brick beyond the range of its structural stability under the load conditions to which it is subjected. Frequently this heating is accentuated by pressure of gases on the fire face.

Furnace-Gas Pressure

116 All firebrick are highly porous and, subjected to hot gases under pressure, are heated by convection from these gases as well as by the normal process of transmission. Howe and Phelps [29] experimenting with a tile wall 9 in. in thickness found that with a furnace temperature of 2444 deg. fahr. (1340 deg. cent.), a change in pressure of furnace gases of from −0.30 to +0.05 in. of water raised the temperature within the brick at a depth of 3 in. from the fire face from 1328 deg. fahr. (720 deg. cent.) to 1742 deg. fahr. (950 deg. cent.), and a further increase of furnace gas pressure to +1.00 in. of water brought an increase of temperature to 1904 deg. fahr. (1040 deg. cent.) at the same position. These figures clearly indicate the importance, from the standpoint of the refractory, of maintaining the higher pressure on the cooler face of the wall.

Slag Erosion

117 All commercial fuels, with the possible exception of gas, have mineral residues which are incombustible and, at temperatnres sufficiently high, form eutectics with the constituents of firebrick. These eutectics have melting points materially lower than those of the original brick. In general such action is very greatly accelerated in the presence of a reducing atmosphere, such as sometimes occurs along the walls immediately above stoker retorts unless provisions are made for admitting air at this point. The slag that is formed, if sufficiently fluid, will not only flow down the face of the wall but will be drawn into the pores of the brick. Usually, the plastic clay with which the large particles of the brick are bonded together, having the greater percentage of impurities, will most readily combine with the slag and, becoming fluid, carry away the more resistant flint particles by mechanical flotation. Fig. 13 gives an illustration of this action in which it will be seen that some of the particles of flint clay are completely immersed in

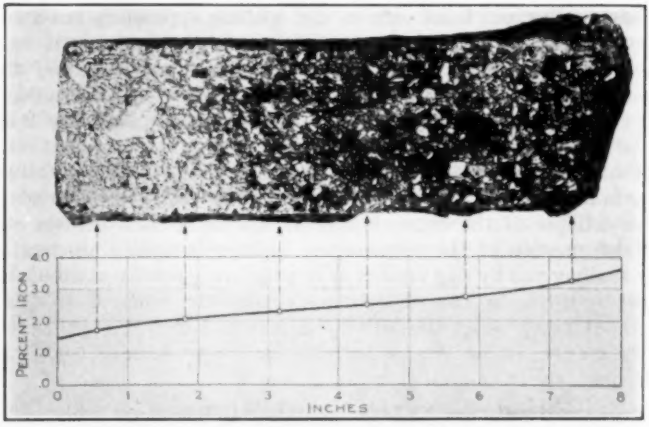

FIG. 13 SLAG PENETRATION, FIRECLAY BRICK, STOKER-FIRED FURNACE

the slag. (The figure also shows one of the common causes of lack of uniformity in firebrick. The piece of flint at the left is 1 in. in diameter.) This brick, which was removed from the wall shown in Fig. 12 after about 2000 hours of service, shows absorption of

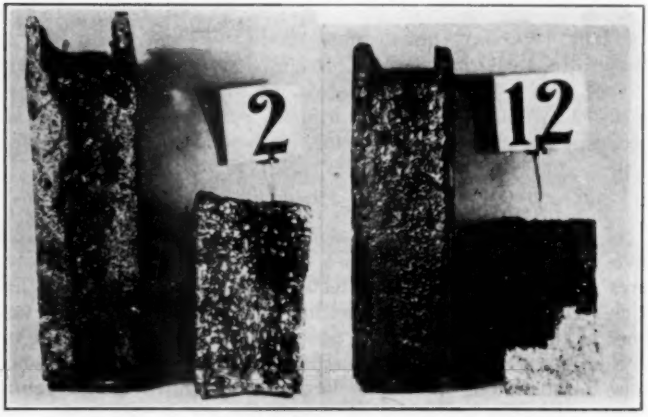

FIG. 14 SLAG TESTS, EFFECT OF INDIANA COAL SLAG

iron to considerable depth. The variation of iron content through the brick is given by the curve in the lower part of the figure. At the heated end the iron appears to have been reduced to the metallic form. With some materials, the slag retains a very high viscosity until a critical temperature is reached so that a furnace may be operated up to a certain rate of capacity without serious deteriora-

FIG. 15 SLAG TESTS, EFFECT OF ILLINOIS AND INDIANA COAL SLAG

tion but at any higher rate, even a very few per cent increase, erosion occurs with extreme rapidity.

118 The resistance of firebrick to the action of coal-ash slag will be governed, not only by the temperature and atmosphere to which it is subjected in the furnace and the chemical composition of the brick in relation to that of the slag but also, very markedly,

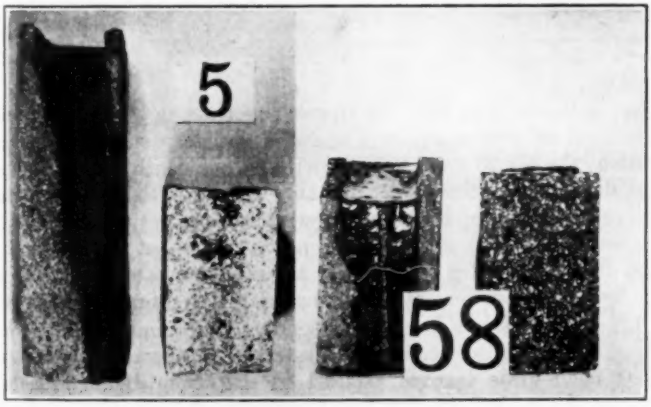

FIG. 16 SLAG TESTS, EFFECT OF ILLINOIS AND INDIANA COAL SLAG

by the texture of the brick. Figs. 14, 15, and 16 show specimens of a number of different brands of brick after subjection to a laboratory slag test[22] at 2700 deg. fahr. (1482 deg. cent.). As grouped in the figures, the specimen at the left presents a 2½ by 9-in. face of the brick over which the slag has been permitted to flow after melting in the small trough in the top of the brick,

and, at the right, one of the brick is shown broken in two to show the nature of slag action and depth of penetration. The slag used on these tests was from an Indiana coal for specimens Nos. 2, 5, 7, and 12, and from an Illinois coal for specimens Nos. 4, 23, and 58. Analyses of the two slags are given in Table 6. The composition of the different brick is given in Table 5.

119 Brick No. 2 represents the same brand of hand-made Maryland clay brick referred to by the same number in Fig. 10. The slag is shown to have penetrated throughout the structure of the brick; in fact, comparatively little slag has flowed over the edge of the trough in the top of the brick, the greater part having been absorbed as rapidly as melting occurred. There is also very noticeable swelling as a result of this slag absorption.

120 Specimen No. 7 is a Missouri clay brick made by the stiff-mud process, and No. 23 is a brick of identical composition made by the dry-press process. The photograph for No. 7 shows the

TABLE 6 ANALYSIS OF COAL SLAG

Coal	No. 1 Indiana No. 5 Vein	No. 2 Illinois Kincaid
Slag	Per cent	Per cent
Silica	42	43
Alumina	15	17
Ferric oxide	33	27
Titania	..	1
Lime	3	7
Magnesia	Trace	1
Alkalis	7	4

effect of the Indiana slag; the Illinois slag tried on this same brick evidenced no greater effect. The No. 7 specimen shows no penetration; the No. 23 specimen is penetrated to a depth of about $\frac{3}{4}$ in. The dry-press specimen was probably burned initially to slightly higher temperature; it shows somewhat lower porosity, about 14.5 per cent; the stiff-mud brick has a porosity of about 17.3 per cent. The difference in slag resistance is apparently due to the effect of the wet process in cementing the materials into a firmer structure.

121 Fig. 17 shows brick corresponding to specimen No. 7 after about 5600 hr. of service in the high-temperature zone of a forced-draft chain-grate furnace exposed to slag from Indiana coal.* During this period the slag was frequently of lower iron content and consequently of less severe chemical action than that represented by the analysis of Table 6. However, the condition of this brick may be regarded as confirming, at least in a general way, the deductions of the slag test. While erosion has occurred slag penetration has been almost negligible. Chemical analysis shows

* The laminations of structure may partly account for the spalling tendency sometimes shown by this brick.

only slight increase of iron content, and this difference may possibly have been due to variation in composition of the original brick.

122 Specimen No. 12 is another Missouri brick but of rather high alumina content. The material of this brick is very porous and at the temperature of the tests seems to absorb the slag and actually combine with it. In the case of this brick, however, there is no noticeable swelling as the result of slag absorption such as occurred in the case of specimen No. 2.

123 Specimen No. 4 is chiefly of interest as showing the effects of a laminated structure. The actual erosion was very slight but the slag flowing between the laminations penetrated deeply into the brick.

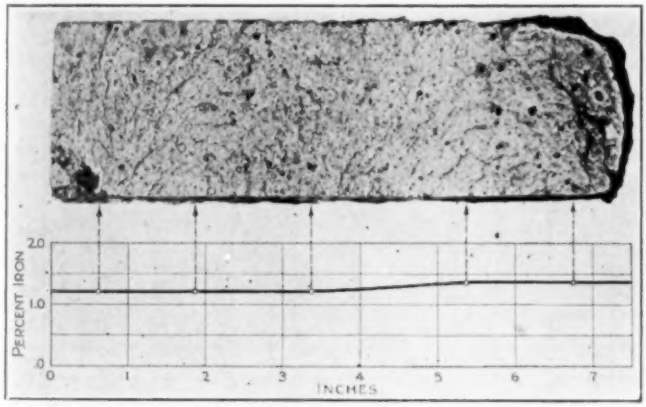

FIG. 17 SLAG PENETRATION, FIRECLAY BRICK, STOKER-FIRED FURNACE

124 Specimens Nos. 5 and 58 are shown as another instance of difference in slag resistance due to difference in manufacture. Specimen No. 5 was made by the semi-dry-press process of medium grind and burned to an average porosity of about 22 per cent. In specimen No. 58, brick of the same composition and manufacturing process were made with finer grind and given a much harder burn, the porosity being reduced to about 13 per cent. Specimen No. 5 shows penetration of Indiana slag to a depth of about $\frac{3}{4}$ in. at the fire face. In addition, the brick structure is evidently permeated with cracks by following which the slag was enabled to find its way through the brick. One of these cracks, which did not reach to the top of the brick, is shown at about the center of the specimen at the right. Above this crack can be seen slag which has penetrated from the top of the brick, evidently following some such crack. Specimen No. 58, on the other hand, shows only very slight erosion with penetration to a depth of perhaps $\frac{1}{8}$ in. The Illinois

slag was used in the test of No. 58 but there is insufficient difference in the fluxing action of the two slags to account for any marked difference in penetration.

125 The chemical activity between coal-ash slag and firebrick is not readily predictable from analysis alone on account of the indefinite nature of eutectics formed. Small variation in basic oxides in either brick or slag may have marked effect upon the melting temperature of the two materials in contact. For reasons of such differences in composition the chemical activities of different slags as affecting different brick will show no dependable relation but vary widely and irregularly. Within the past three years results of slag fusion tests have been published by members of the Mellon Institute Industrial Fellowship.[27] The Detroit Edison Company,[28] and by J. J. Brennan [29] of the Northern States Power

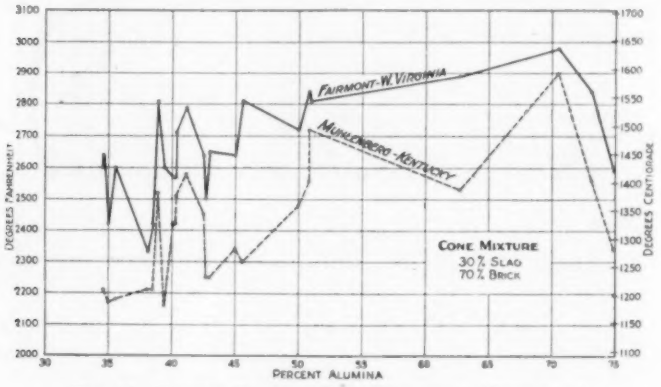

FIG. 18 FUSION TESTS OF SLAG MIXTURES (BRENNAN)

Company. In these tests ground brick of different brands were separately mixed in definite proportions with ground slag from various steel-plant and steam-boiler furnaces using different grades of coal. Test cones were formed of these different mixtures and fusion points determined. Some of Brennan's data are presented in Fig. 18. On this chart fusion temperatures are shown for two series of test-cone mixtures of ground firebrick and coal ash combined in the ratio of 70 per cent firebrick, 30 per cent ash. The two slags used are ash from Fairmont, West Virginia, coal and from Muhlenberg County, Kentucky, coal. The fusion points of the different cones appear in horizontal sequence, in order and composition corresponding to the alumina content. While some of the test cones show fairly consistent relationship in the two curves, others vary rather irregularly.

126 The character of the furnace atmosphere, as pointed out, has marked effect upon the fusion temperature and resultant flux-

ing action of slags. The extreme rate of erosion shown in Fig. 19 resulting from 14½ days operation can only be accounted for as due to an unusual reducing atmosphere in contact with the surface affected. In a number of plants, both oil and coal burning, materially increased life of refractories has been obtained by admitting air to the furnace at frequent openings through the wall in the zone of most rapid deterioration. It would appear probable that the viscosity of the slag is increased not only by the lower temperature but also by the somewhat oxidizing atmosphere produced by the ventilating air.

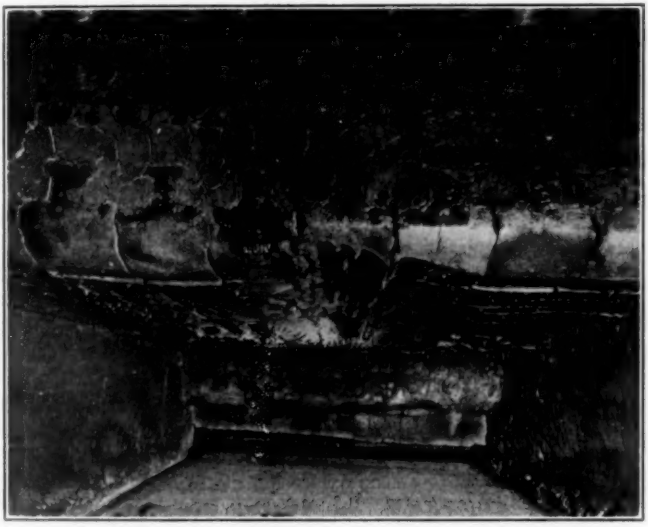

FIG. 19 EROSION OF ARCH RESULTING FROM 14½ DAYS OPERATION

CLINKER ADHESION

127 The ash from many of the higher grades of coal is not sufficiently fluid at the usual furnace temperatures to cause difficulty from fluxing of the brickwork. On the other hand there is practically no coal, the ash from which does not become sufficiently plastic at such temperatures to adhere to the ordinary refractory wall. With some coals, if the face of the wall is maintained below about 2200 deg. fahr. (1204 deg. cent.), the ash does not adhere strongly and may be removed without damage to the brickwork. With other coals, the presence of a thoroughly oxidizing atmosphere at ordinary furnace temperatures will prevent very strong adhesion to the brickwork. With still other coals, however, a hard

and very adherent clinker will form at temperatures as low as 2000 deg. fahr. (1093 deg. cent.). When these coals are stoker fired the avoidance of clinkering difficulty requires special cooling at the slag line as it is necessary that the face of the refractory be maintaned at a temperature materially below the average of the furnace. With pulverized-coal firing it is sometimes possible to raise the furnace temperature above the softening point of the ash and so avoid excessive clinker formation and at the same time escape serious slag erosion.

128 When it is necessary to chip away the slag or clinker, it is practically impossible to avoid some fracture of the brickwork. The extent of the damage will, of course, depend upon the depth of penetration and the strength of the brick. In general the same qualities of density and hardness of structure which make for resistance to the fluxing action of slag will also tend to minimize clinker adhesion.

Thermal Expansion and Mechanical Stress

129 Fracture under mechanical stress is a far more common failure than is generally supposed. Practically all refractory materials are subject to thermal expansion and, if provision is not made for this expansion, fracture of some kind must necessarily result. Expansion stresses have been a frequent cause of tile breakage in suspended arches. With increasing dimensions, this type of failure has extended to bridgewalls and to side walls.

130 In the comparatively small boiler furnaces of a few years ago, raw clay was used almost exclusively for mortar, and joints were rarely less than $\frac{3}{16}$ in. in thickness and often greater than $\frac{1}{4}$ in. The greater part of the expansion of the wall was taken up by shrinkage in the joints and, although these walls were commonly distorted and cracked, actual failure of the lining was only an occasional occurrence. With the thin joints and mortar containing a high percentage of calcined material now commonly used, the structure is too compact to absorb expansion movements without strain. The recent pulverized-coal furnaces with ventilated walls anchored to outer steel framework have been particularly subject to expansion failures. In these furnaces the refractories have been stressed under tension, bending, and shear. The strength of firebrick in bending is about one-half to one-third the strength in compression, and in shear is probably less than one-third, and, in tension, less than one-fifth the compression strength. A hard-burned firebrick may show tensile strength as high as 500 to 700 lb. per sq. in.; the softer-burned brick of good quality will fail under tension as low as 130 to 175 lb. per sq. in.

131 The average firebrick under loads and temperatures such as occur in boiler settings will expand about 0.6 per cent. Some

brick have materially lower expansion and others will expand as high as one per cent. The curves of Figs. 10 and 11 show thermal expansions under a load of 25 lb. per sq. in. and in connection with Fig. 11 it has been pointed out that, for brick of a particular composition, a higher temperature of burn, with its greater conversion of uncombined silica to the cristobalite and tridymite forms and other changes, gave a lower maximum of thermal expansion.

132 Houldsworth and Cobb,[29] at the University of Leeds, England, have made very careful determinations of the thermal or temporary expansion characteristics of a number of different refractory materials. Thermal-expansion determinations were made over the range of 59 to 1832 deg. fahr. (15 to 1000 deg. cent.) on fireclay brick of different alumina-silica ratio, different specimens being previously burned at 1796 deg. fahr. (980 deg. cent.), 2336 deg. fahr. (1280 deg. cent.), 2570 deg. fahr. (1410 deg. cent.), 2786 deg. fahr. (1530 deg. cent.). The brick of relatively high silica content showed a high rate of expansion below 1112 deg. fahr. (600 deg. cent.) for those specimens burned at 2570 deg. fahr. (1410 deg. cent.) and lower temperatures, due to the presence of unconverted quartz and cristobalite. Burning to 2786 deg. fahr. (1530 deg. cent.) however, apparently affected the complete conversion of the quartz to cristobalite, tridymite, and quartz glass and straightened out the expansion characteristic. The clays of approximately kaolin as well as of more highly aluminous composition showed very nearly straight-line expansion characteristics even for the specimen of lowest burn. In the case of each grade of clay, however, the highest rate of thermal expansion was shown for the specimen burned at 2336 deg. fahr. (1280 deg. cent.), the next lowest for the specimen burned at 2570 deg. fahr. (1410 deg. cent.), the next lowest for the specimen burned at 1796 deg. fahr. (980 deg. cent.), and the lowest expansion for the specimen burned 2786 deg. fahr. (1530 deg. cent.). At the highest temperature the combined silica and alumina was doubtless entirely in the form of mullite, the uncombined silica converted to cristobalite, tridymite, and quartz glass, and any uncombined alumina, to corundum.*

133 Houldsworth and Cobb[30] also investigated the effect of varying porosity upon the thermal expansion and found that increase of porosity up to approximately 50 per cent was accompanied by continuous decrease in coefficient of expansion. As these authors have suggested, the lower expansion accompanying the increased porosity resulted from the greater opportunity

* Eventually, of course, all tridymite would be converted to cristobalite if mainained at 2786 deg. fahr. (1530 deg. cent.), but this temperature is so little above the 2678 deg. fahr. (1470 deg. cent.) limit that it would seem probable that, under the conditions of brief laboratory test, tridymite might still be found as an important constituent.

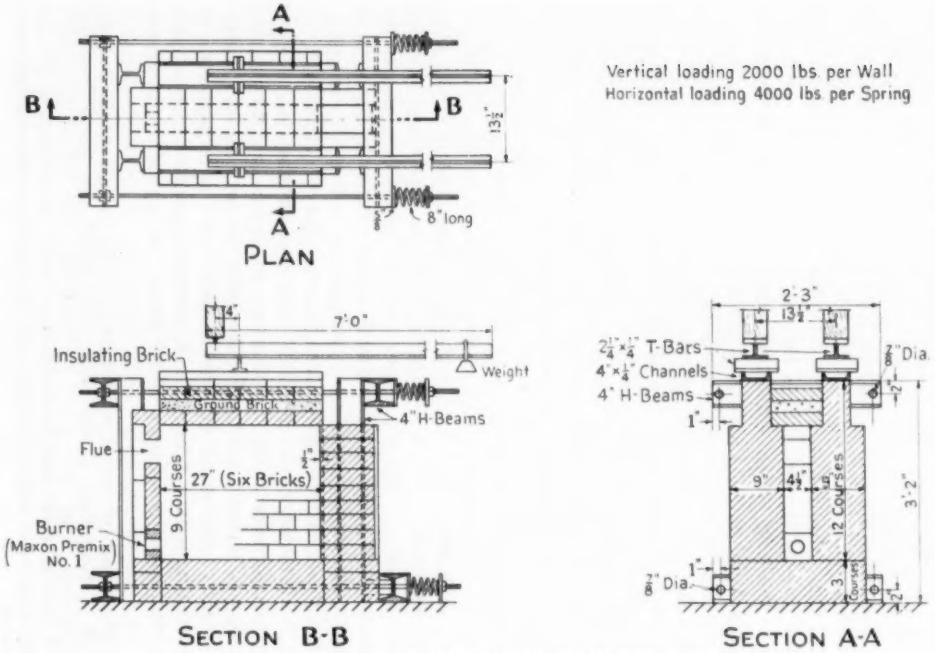

FIG. 20 FURNACE FOR TEST OF WALL CONSTRUCTION

afforded by the larger pore space for absorption of expansion within the brick structure.

134 M. C. Booze and S. M. Phelps [26] at the Mellon Institute have also investigated expansion characteristics of test brick made from certain Pennsylvania clays. They have reported that in most cases reheating the brick for five hours at 2552 deg. fahr. (1400 deg. cent.) increased the coefficient of expansion, although expansion was directly proportional to the silica content. Apparently, in these investigations, the burn at the highest temperature was not sufficient to effect any very large conversion of silica to the low-expansion forms.

Experimental Investigation of Wall Stresses

135 In preparing for the design of boiler settings[22] in which the walls were to be ventilated under forced draft, laboratory studies were made on small walls, simulating the final construction, to gain some measure of the stresses to be encountered and of the expansion provisions which would be required. Two test walls 9 in. thick, about 22½ in. in height, and 27 in. in length were set up about 4½ in. apart to form the opposite sides of a small reverberatory type of furnace which was completely closed except for burner opening at one end and gas outlet about 4½ by 4½ in.

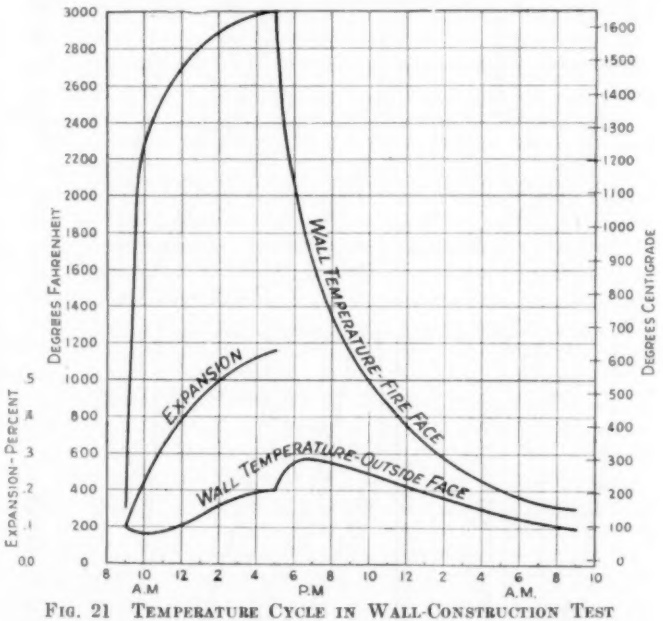

Fig. 21 Temperature Cycle in Wall-Construction Test

above the burner at the same end. Artificial loads were applied both vertically and horizontally and observations made of wall movements and of inside and outside wall temperatures. The general arrangement of the test walls is shown in Fig. 20.

136 Fig. 21 shows a typical day's temperature cycle. The walls were given an outside coating, ¼ in. in thickness, of various mixtures of asbestos and high-temperature cement and, in consequence, the outside temperature is somewhat lower than would obtain with a corresponding wall of bare brick. Conditions of power supply at the laboratory at this time made it impractical to continue tests much beyond five o'clock in the afternoon; in consequence, the duration of the firing period was limited to about

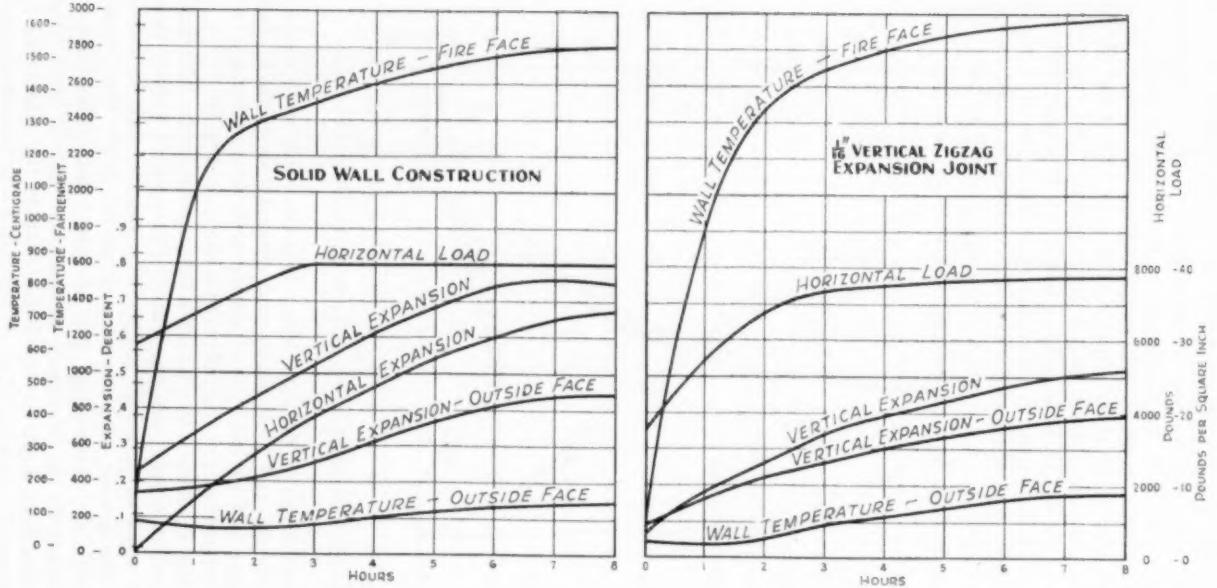

FIG. 22 EXPANSION CHARACTERISTICS IN WALL-CONSTRUCTION TESTS

eight hours. However, this period, while not carrying expansions to full maximum values, gave sufficient results to permit their close approximation.

137 The wall-test furnace was fired by illuminating gas under forced draft, and it will be noted that the temperature rose very rapidly, reaching 2300 deg. fahr. (1260 deg. cent.) in about one hour. On some tests 3000 deg. fahr. (1649 deg. cent.) would be reached in about five hours. At the time the burner was turned on, small blowers were started, one playing on the outside of either wall, blowing ventilating air at a velocity of about 900 ft. per min. along the face of the wall. At five o'clock in the afternoon the burner was turned off; the fire-face temperature then dropped rapidly. At the same time, as the ventilating blowers were also shut down, there was an immediate rise of temperature on the outside face of the walls, generally reaching about 500 deg. fahr. (260 deg. cent.). For the particular brick shown it will be noted that expansion continued from about 0.10 per cent, corresponding to the condition of the wall at nine o'clock in the morning, to about 0.58 per cent at the close of the firing run. The maximum expansion for this particular brick under a 25-lb. load was about 0.60 per cent. From the trend of the expansion curve, this value would probably have been reached and slightly exceeded in about two or three hours. Fig. 22 shows the complete data for two walls set with other brick, one of which was provided with an expansion joint about mid-length of the wall and the other without special provision to take up expansion. It will be noted that, in the wall without expansion joint, horizontal expansion took place at fairly uniform rate in spite of a restraining load of about 8000 lb. per wall, equivalent to approximately 40 lb. per sq. in. of vertical-wall cross-section, and that, in the wall with $\frac{1}{16}$-in. zigzag expansion joint provided, the full stress of 8000 lb. was required to confine the horizontal movement of the wall to this joint. In some of the other walls of this type the horizontal stress reached practically 10,000 lb., equivalent to about 50 lb. per sq. in. of vertical-wall cross-section. The stresses indicated are those required to restrain a wall of the given dimensions under a vertical load of only about 8 lb. per sq. in., which was the maximum that could be conveniently applied in the laboratory. It is probable that under heavier loads these stresses would be somewhat increased.

138 The brick used in the wall-construction tests of Fig. 22 were of lighter burn than in the case of the test represented by Fig. 21 and accordingly were more responsive to difference in applied vertical load. The maximum expansion shown by the brick used in wall $5B_r$ under 25 lb. load, was about 0.60 per cent; under the 8-lb. load of the wall test this value was exceeded after about four hours, the final maximum expansion having been about 0.76 per cent. The brick used in wall 6B, under 25 lb. load, expanded about 0.40 per cent; in the wall test the maximum expansion

reached was about 0.52 per cent, and somewhat higher expansion would doubtless have been obtained on a longer run. The vertical load was applied over the longitudinal center line of the wall and so permitted some self-adjustment of the wall and a smaller movement of the outside face.

Spalling

139 The term "spalling" is commonly applied to any type of fracture of firebrick or other refractory structure, resulting from the application of heat, from which pieces of refractory are dislodged to fall into the furnace. In many instances the fracture, while occasioned by heat, is primarily the result of faulty construction or failure in the design to take account of expansion movements and stresses which occur. For the sake of clarity, the term "spalling" should be applied to the refractory breakage which results from thermal stresses, both originating and applied, essentially within the individual refractory. Fractures resulting from stresses which are in anywise dependent upon the existence of solid materials external to the individual refractory should be charged to design or construction and not to lack of spalling resistance on the part of the refractory. Accepting this definition, spalling is the result of thermal shock more or less repeated.

140 As the temperature of the furnace is increased, the exposed face of the refractory is heated comparatively rapidly but, owing to the relatively low conductivity of most refractory materials, the temperature of the interior rises only very slowly. As the furnace is cooled the reverse action takes place and the temperature of the face of the refractory falls below that of the interior. The effect is to subject the heated end of the brick alternately to compressive and tensile stresses. The exposed end will expand until its softening temperature is reached and this expansion will continue progressively through the brick toward the cooler end following the rise of temperature. On cooling, the furnace end of the brick will tend to contract and this movement will be resisted by the hotter interior. As pointed out by F. H. Norton [24] in A General Theory of Spalling, the resulting shearing stress will be applied along lines of equal temperature.

141 The mechanically atomized oil-burning furnace has introduced the most serious problem of spalling. The fire-face temperature changes in the gas-fired laboratory furnace are shown in Fig. 21. In this furnace the surface temperatures were raised at a rate as high as 2000 deg. fahr. (1093 deg. cent.) per hour and cooled at 900 deg. fahr. (482 deg. cent.) per hour. While actual data for oil-burning furnaces are not available, exposure of the boiler heating surface would probably produce a much greater rate of temperature drop. Observations made in the 1924 survey [25] under service conditions in solid-wall coal-burning furnaces have shown that, at positions within the brick ½ in. from the fire face

temperature rises as rapid as 1250 to 1275 deg. fahr. (677 to 691 deg. cent.) per hour occur with underfeed or traveling-grate types of stoker and at 1300 deg. fahr. (704 deg. cent.) per hour, 1 in. from the fire face, with pulverized coal. Wall cooling in these

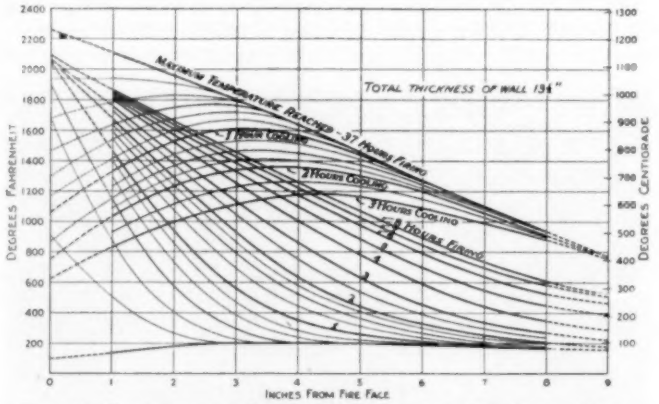

FIG. 23 TEMPERATURE CHANGE IN FIREBRICK, PULVERIZED-COAL-BURNING FURNACE

furnaces takes place at a much lower rate. For the underfeed-stoker furnace a decrease of 375 deg. fahr. (191 deg. cent.) per hour at a depth of ½ in. from the fire face was recorded following a reduction in output from 150 to 50 per cent of rated capacity. For

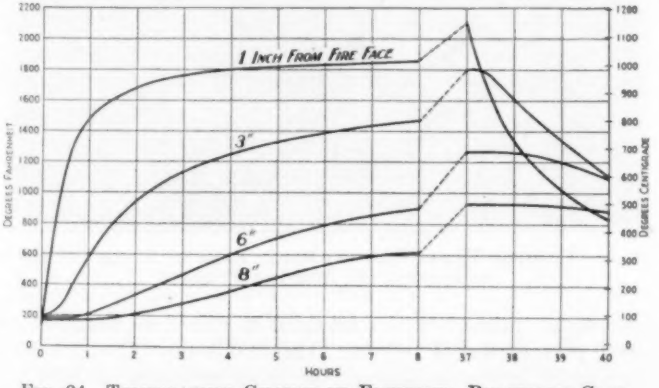

FIG. 24 TEMPERATURE CHANGE IN FIREBRICK, PULVERIZED-COAL-BURNING FURNACE

the traveling-grate furnace a drop of 550 deg. fahr. (288 deg. cent.) per hour was obtained when reducing the rate of operation from about 240 per cent to practically 0. With pulverized-coal firing, the temperature 1 in. from the fire face in the solid brick portion

of a partly water-cooled furnace was found to drop at the rate of 770 deg. fahr. (410 deg. cent.) per hour on abrupt shutting off of burners with damper left open, giving 0.3 in. furnace draft after operating at 200 per cent of rating.

142 In many instances the decrease of fire-face temperature was accompanied by a continued rise of temperature a few inches within the brick. Figs. 23 and 24 show firing-up and cooling tem-

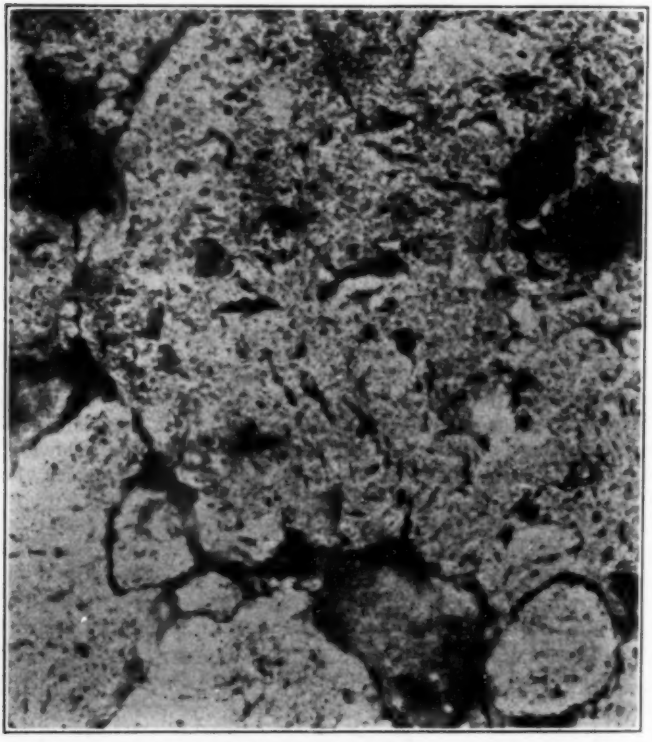

FIG. 25 VITRIFIED END OF FIRECLAY BRICK, REMOVED AFTER 1150 HOURS OF SERVICE, OIL-BURNING FURNACE

peratures as observed in the solid firebrick wall of the partly water-cooled pulverized-coal furnace. The first series of curves show temperature gradients through the brick measured at intervals, first as the boiler was brought from the cold condition to about 200 per cent of the rated capacity, and second, as cooled from this rate after a total of 37 hours operation, the damper being left open giving a furnace draft of about 0.3 in. The other series of curves show the temperature variations plotted for the individual positions in the brick. The rise of temperature takes place progres-

sively as the heat is gradually transmitted through the brick. In cooling, the fire-face temperature falls most rapidly, the temperature 1 in. from the fire face falling below that of the point 3 in. from the fire face about twenty minutes after the cooling begins and below that of the point 6 in. from the fire face at the end of a little more than an hour. The 3-in. point is hardly affected at the end of five minutes and the 6-in. point remains constant for nearly an hour. It will be readily appreciated that the more rapid

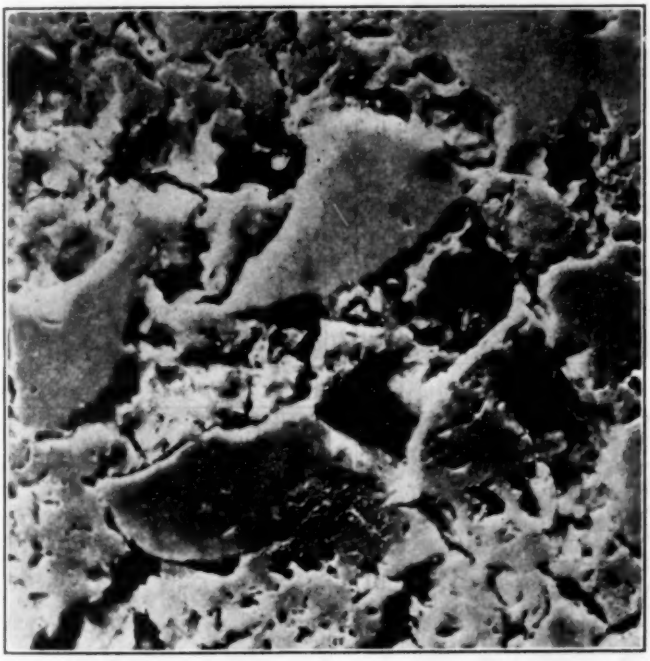

FIG. 26 STRUCTURE OF KAOLIN BRICK. KILN BURN 3000 DEG. FAHR. (1649 DEG. CENT.)

cooling of the exposed end of the brick causes very serious stresses. At temperatures at which the brick are moderately plastic these stresses are absorbed without damage to the brick, but on cooling to lower temperatures, unless the brick structure possesses sufficient permanent elasticity, fracture results. The curves of temperature rise, Figs. 23 and 24, incidentally give an indication of the time through which boiler furnace walls continue to absorb heat.

143 From these curves, it is evident that at a point within the brick 3 or 4 in. from the fire face, temperature changes accompanying ordinary fluctuations in load are inappreciable. The gradual

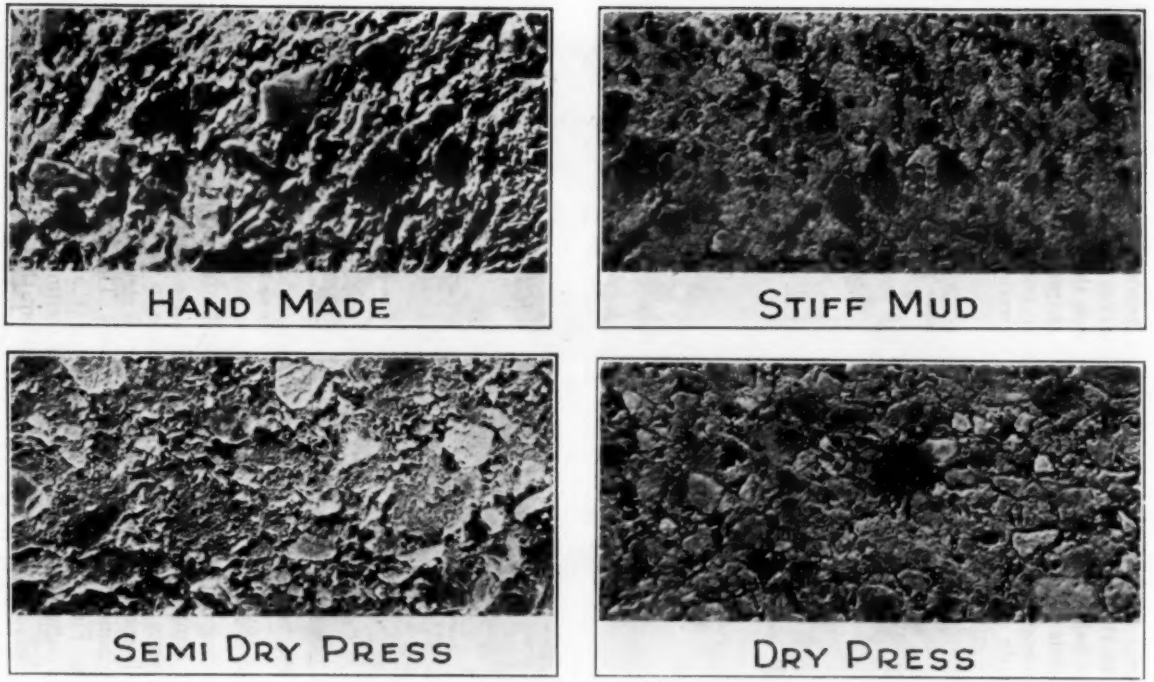

FIG. 27 STRUCTURE OF FIRECLAY BRICK

accumulation of clinker which commonly occurs in the stoker-fired furnace with coals of less fusible ash affords a protective coating which frequently moves this point to the face of the wall so that the brick are practically free from thermal shock. If, however, the normal rate of operation is materially increased, or if coal of lower fusing ash is burned, the coating may be completely fluxed leaving the brick unprotected, and spalling difficulties may then appear in walls which previously had been free from this form of deterioration. The absence of protective coating frequently makes spalling a serious problem in furnaces burning midwestern coals of low fusion ash when operated at relatively high rates. Fig. 19 shows the interior of a boiler furnace, burning coal of this character. The brickwork is particularly clean of slag and, in addition to the erosion, spalling has begun in many places on the face of the arch. In fact, it is entirely possible that much of the apparent erosion is primarily occasioned by spalling.

144 Vitrification has frequently been assigned as the important factor in failure by spalling and, in fact, susceptibility to vitrification has been offered as a measure of spalling tendency in fireclay brick. Certain classes of brick show increased tendency to spall following vitrification. Many brick fail by spalling without appreciable evidence of vitrification beyond that imparted by the kiln burn. It would seem probable that the tendency to spall lies rather in the nature of the materials or in the loss of homogeneity of structure resulting from shrinkage or other change coincident with vitrification. Fig. 25 shows a section at five magnifications of the vitrified end of a brick removed after 1150 hours of service from the wall of a mechanically atomizing oil-burning furnace. This brick has shown practical immunity from spalling under conditions where, with the exception of very few brands, the best grades of firebrick have a life of from five to six months. Fig. 26 shows a section, at somewhat greater magnification, of an entirely different type of brick, as originally made. This brick also shows very high resistance to spalling although vitrification is quite apparent throughout the structure. The structures of Fig. 27 are typical of the usual commercial burn of fireclay brick.

145 For moderate conditions of service, with coals which at the temperatures obtained do not give serious trouble from slag or clinkering, the coarse-ground brick of moderately soft burn and open porous structure, such as that of the hand-made brick of Fig. 27, has frequently shown the highest resistance to spalling. The temperature is not sufficiently high to cause serious shrinkage so that homogeneity is fairly well maintained. The porous structure apparently distributes the stresses and permits a large part of the expansion to be taken up within the brick. This flexibility of the structure, however, is dependent upon the nature and condition of the materials used even more than upon the apparent texture, and mere visible inspection can afford no criterion of the actual spalling resistance.

146 In 1924, F. H. Norton [24] made a careful study of the mechanism of spalling and reported the important physical properties bearing upon this type of failure as coefficient of expansion, flexibility in shear, and diffusivity. He measured these properties quantitatively for a number of types of refractories and developed a mathematical theory for their relations to spalling resistance which, compared with results of actual spalling tests on the brick, showed very consistent agreement. Norton's theory may be summarized as follows: The spalling resistance of the brick is proportional to the product of the square root of the diffusivity and the maximum shearing strain divided by the coefficient of expansion. M. C. Booze and S. M. Phelps [25] made a simultaneous but independent study of the subject, and, while they have not placed their theory in formula form, arrived at the same conclusions as Norton, that spalling resistance was proportional directly to conductivity and flexibility and inversely to specific heat and coefficient of expansion. The conclusion of both investigations was that flexibility was the most important of these properties.

147 For the ordinary fireclay materials, thermal conductivity at high temperatures is not susceptible of very wide variation; specific heat, which is the other factor in diffusivity, is practically independent of manufacturing treatment. The normal range of thermal conductivity and specific heat in fireclay brick is given

TABLE 7 THERMAL CONDUCTIVITY AND SPECIFIC HEAT OF FIRECLAY BRICK

Temperature range	Thermal conductivity		Specific heat
	Gram-calories per sec.	B.t.u. per 24 hr.	
77-932 deg. fahr. 25-500 deg. cent.	0.0021 ± .0003	146 ± 21	0.223 ± .010
77-1832 deg. fahr. 25-1000 deg. cent.	0.0025 ± .0003	174 ± 21	0.259 ± .010
77-2732 deg. fahr. 25-1500 deg. cent.	0.0030 ± .0004	209 ± 28	0.300 ± .010

in Table 7. The coefficient of expansion has been shown to decrease for certain brick with increased burn beyond 2336 deg. fahr. (1280 deg. cent.). Flexibility is perhaps the property most subject to development in manufacture.

148 The important consideration, however, is not so much the measure or relation of these properties in the manufactured brick but the extent to which they may appear in the brick in service. When the pores of the brick have become filled with slag or when permanent volume change of either expansion or contraction has occurred, or if there has been further vitrification, it is impossible that the physical characteristics of the brick should remain unaffected. Many brick which show excellent resistance to spalling under any of the present forms of laboratory test, when applied to actual service, undergo structural change and spall very badly. Fig. 28 shows an instance of side-wall failure in a mechanically atomized oil-burning furnace. The brick used in this furnace, under

test, showed very creditable spalling resistance. At the temperatures of the furnace, shrinkage and vitrification occurred which resulted in radical change of structure, the vitrification being

FIG. 28 SIDE-WALL SPALLING, OIL-BURNING FURNACE

doubtless aided by the absorption of slag from the oil. Fig. 29 shows a number of brick at various stages of spalling, from oil-burning service, which present many of the characteristics distinctive of that type of failure. Specimens Nos. 22, 48, and 65 show fracture across the face as a result of rapid contraction from the

sudden cooling to which the oil-burning furnace is commonly subjected. Fusion will sometimes greatly increase the strength of the exposed face of the brick so that fracture will occur a short distance back from the face. In this case the end of the brick may shear off in a thin layer of fairly uniform thickness as shown in specimen No. 56. Spalling failures in coal-burning furnaces occur in a generally similar manner; the brick sometimes fractures across the face and in other instances where the slag is of sufficient strength the end of the brick to a depth of one or two inches is sheared off intact.

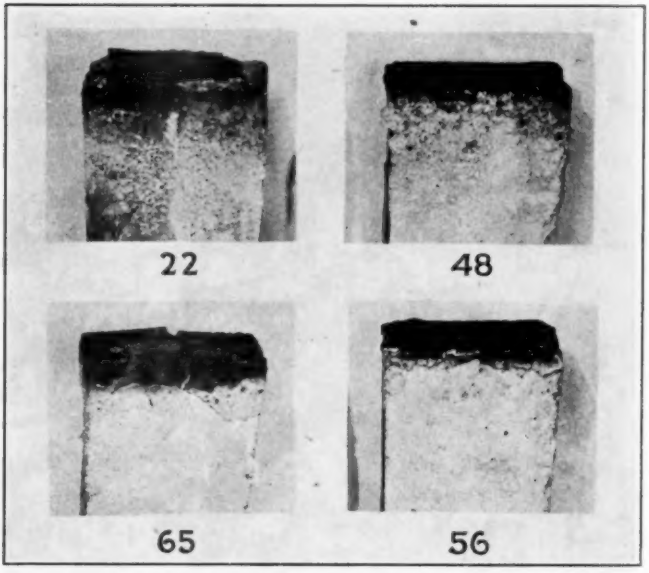

FIG. 29 SPALLING OF FIREBRICK, OIL-BURNING FURNACE

149 Fig. 30 shows results of a series of spalling tests on brick of various brands from different parts of the country. Some of these are presented chiefly to indicate the variation in spalling resistance shown by brick of the same manufacture and others to show the degree of uniformity which may be had in brick of other brands. The spalling tests referred to were conducted by heating one end of the brick to 2900 deg. fahr. (1593 deg. cent.) and cooling in air. The brick were not previously reheated. The majority of the brick shown are handmade. Brick No. 23 was made by the dry-press process, and No. 29 by the semi-dry press. Brick Nos. 12 and 20 are special brick of relatively high alumina content and high-temperature burn. For many grades of clay

brick, of silica content higher than the kaolin ratio, higher burning temperature causes vitrification with increased density and increased susceptibility to spalling. With the more aluminous brick, however, those containing around 50 per cent of alumina or higher, the increased temperature of burn, at least within the range from cone 14 [about 2570 deg. fahr. (1410 deg. cent.)] to cone 18 [about 2714 deg. fahr. (1490 deg. cent.)] is accompanied by increased spalling resistance. One such brick, for example, had a resistance equivalent to about 9 cycles of air cooling from 2900 deg. fahr. (1593 deg. cent.) when burned at cone 14, and an average of 20 such cycles when burned at cone 18. Other brick of

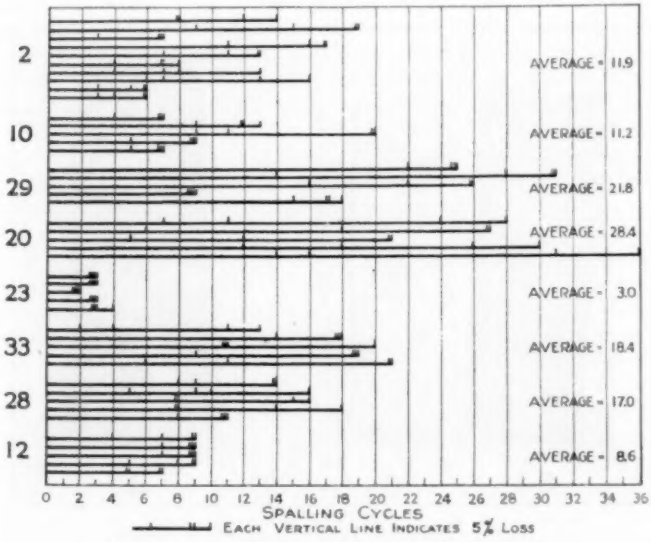

FIG. 30 AIR SPALLING TESTS, 2900 DEG. FAHR.

this same general composition have shown even more marked change of spalling tendency accompanying the corresponding change in burning temperature. In the case of the aluminous brick this decreased spalling tendency is probably due to greater conversion to mullite coupled with a smaller shrinkage at the temperature of the test.

STRUCTURAL CHANGE

150 Some of the less aluminous fireclays show a relatively constant "permanent" volume over a fairly wide temperature range, and may be satisfactorily used at temperatures higher than that of the initial burn. Others are subject to permanent volume change, either expansion or shrinkage, on comparatively small

change of temperature. By combining clays of opposing characteristics it is sometimes possible to produce a brick of much more nearly stable volume than could be had with either clay alone.

151 Stresses due to restriction of thermal expansion not only tend to a compression of the heated end of the brick but accelerate vitrification[31] which, for some materials, has further detrimental effect upon the spalling resistance. Shrinkage opens the joints and so exposes the sides of the brick to furnace heat and contact of gases, increasing the tendency to vitrification and carrying the shrinkage further into the body of the brick.

152 Any permanent change of volume after a refractory is placed in service necessarily occurs chiefly at the exposed end, and

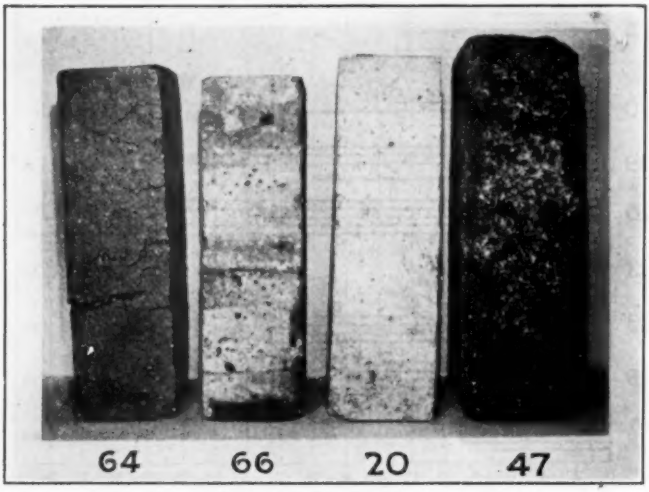

FIG. 31 REHEAT SPECIMENS, FIRECLAY BRICK

largely on that account, destroys the homogeneity of structure, tending to cause weakness along the plane where this structural change occurs. Where the volume change is slight or where the gas causing an expansion forms very slowly and does not produce mechanical rupture of the structure in escaping, there may be no material damage to the brick. Volume changes of 3 per cent or more will usually cause serious injury. Fig. 31 shows distortions of structure which sometimes occur from reheating at 2732 deg. fahr. (1500 deg. cent.), specimens Nos. 64 and 66 shrinking in linear dimensions about 2 and 5 per cent, respectively, No. 20 being practically unaffected, and No. 47 expanding about 5 per cent in linear dimensions.

153 Changes of crystalline structure have long been recognized as an important factor in the properties of silica refractories and

porcelains. The petrographic microscope has been extensively employed in the study of these materials and, especially in the case of porcelain, the present development of the industry has, to a large extent, been made possible by the knowledge gained through microscopic study of structure. Recently the suggestion has been made that the qualities of fireclay brick might also be greatly benefited by similar study. Work of this kind has been undertaken by a few investigators in this country and in England, but such studies have not proceeded far enough to have arrived at thoroughly definite conclusions.

154 Investigations by Hewitt Wilson [22] and others have shown that in some refractory materials the conversion of alumina and silica to mullite results in a weakening of structure. This has probably been due to the concentration of impurities in the uncombined silica which occurs as the crystallization of mullite takes place. From the nature of the mullite crystal and from its known influence upon the strength of porcelain, it would be expected that, freed from the effects of impurity in the bond, it would give a much tougher and more flexible structure in firebrick. Unless counteracted by other influences, the conversion of quartz to tridymite should also markedly improve the resistance to temperature change.

155 Dr. Herbert Insley has made petrographic analyses of a number of brick taken from actual service, corresponding to the specimens shown in Figs. 25, 26, and 29. Brick Nos. 13 and 35 have had exceptionally high spalling resistance in service and brick No. 35 fails almost at once in the 2900 deg. fahr. (1593 deg. cent.) air spalling test. The petrographic analysis throws rather instructive light on the characteristics of these brick. Brick No. 13 in the unused state, shown in micrograph Fig. 32, was found to be composed almost wholly of well-developed mullite crystals and glass, and, as might have been anticipated from the temperature at which the brick are burned, service in the oil-burning furnace made no appreciable change in the crystalline structure. Brick No. 35, Figs. 33, 34, and 35, is of very high quartz content and in the unused state the quartz grains showed but little evidence of solution and but slight conversion to cristobalite; there were, however, a few aggregates of tridymite representing preburned material used in the manufacture of the brick; mullite was abundant and sometimes well developed. After service in the furnace, the quartz grains show greater conversion to tridymite or cristobalite at $1\frac{1}{2}$ in. from the fire face, and at about $\frac{3}{16}$ in. from the fire face the conversion has been largely effected. Nearer the fire face, quartz and tridymite show evidence of much solution, and slag has filled the pore channels and solution spaces in the brick. There was, however, no evidence of slag penetration at a depth of $\frac{3}{16}$ in. It would seem that, in service, the conversions of quartz to tridymite or cristobalite which take place at the heated end of this

brick must occur under such conditions of plasticity of structure that no serious stress results. The failure in the spalling test is, of course, due to the high content of quartz in the unused brick.

156 Of the other specimens examined, Nos. 22, 48, and 56 were normal fireclay brick, and No. 65 contained a high percentage of diaspore. The clay brick, unused, show about 10 to 25 per cent of the amount of quartz observed in No. 35. The quartz grains have very litle solution; mullite is but slightly developed in No. 56, appears as very minute needles in No. 48, and is practically absent from No. 22. There is much amorphous material in all three of the clay brick, particularly in No. 22. The unused diaspore brick was found to be composed largely of minute rounded grains, probably incipient mullite crystals, embedded in a relatively small amount of glass, and rounded grains of corundum. The changes effected at the heated ends of these brick, in service, are indicated by the micrographs of Figs. 36 to 41, inclusive. At 1½ in. from the fire face, the clay brick show the beginning of mullite development and materially increased solution of the quartz grains, and the diaspore brick show an increase in size of mullite crystals. Nearer the fire face the quartz in the clay brick goes into solution and there is increasing development of mullite in both classes of brick.

157 Stability of crystalline structure under service conditions is of great value in the durability of the brick. Such stability cannot, of course, offset the effects of deficiencies in mechanical arrangement and sizing of the materials of which the brick is composed. It is entirely possible that for brick of the present usual commercial burn (under 2300 deg. fahr. or 1260 deg. cent.), under temperatures in service but little higher, the arrangement of the crystalline structure of the individual particle is not so important as other factors which may influence the elasticity of the structure of the brick as a whole; in which case, however, knowledge of the microstructure should assist in the interpretation of these other factors. On the other hand, as higher burning temperatures are used and higher temperatures are encountered in service, the crystalline structure of the refractory becomes of very great importance.

TESTS OF REFRACTORIES

158 Tests and specifications for refractories in so far as their application to boiler-furnace requirements is concerned, have, in the past, been based too largely upon manufacturing considerations and knowledge of the properties of materials as gained from laboratory investigation, without the benefit of the coöperation of users. The great need is a form of test which may be interpreted in terms of service performance, or else means by which the present form of test results may be so interpreted and, necessarily, the first requirement is a clear understanding of service conditions.

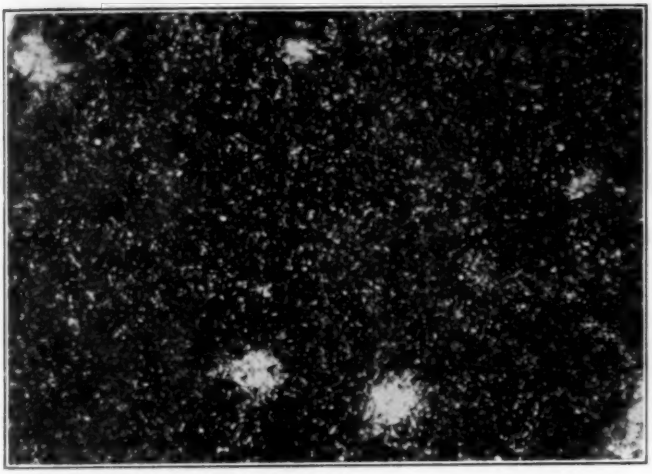

FIG. 32 BRICK NO. 13. UNUSED BRICK. MAGNIFICATION 210×. ORDINARY LIGHT.

Mullite crystals embedded in small amount of glass. No other constituents visible.

FIG. 33 BRICK NO. 35. SECTION ABOUT 1½ IN. FROM SLAG FACE OF USED BRICK. MAGNIFICATION 210×. ORDINARY LIGHT.

Mullite crystals poorly developed — appear as dark areas in photograph. Aggregate of tridymite crystals cover about half the field. Quartz grains show little or no solution.

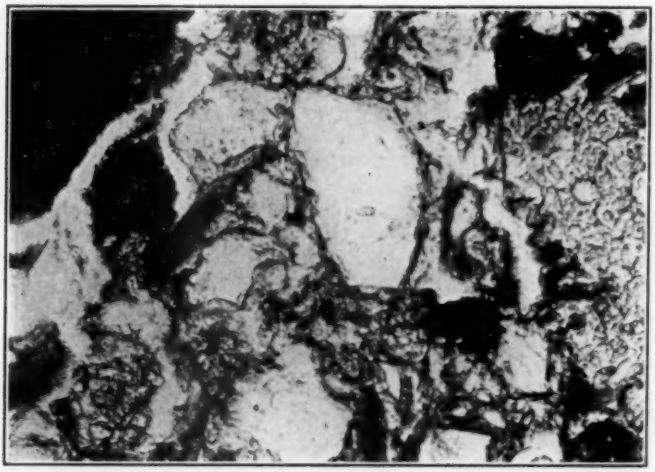

FIG. 34 BRICK NO. 35. UNUSED BRICK. MAGNIFICATION 210×. ORDINARY LIGHT.

Section showing heterogeneous character of brick composition.
Practically no mullite visible in field. Aggregates of tridymite crystals at right, and near lower left-hand corner of field, quartz grains showing practically no solution. Clay matter (dark area at upper left corner) shows very little evidence of conversion to mullite and glass.

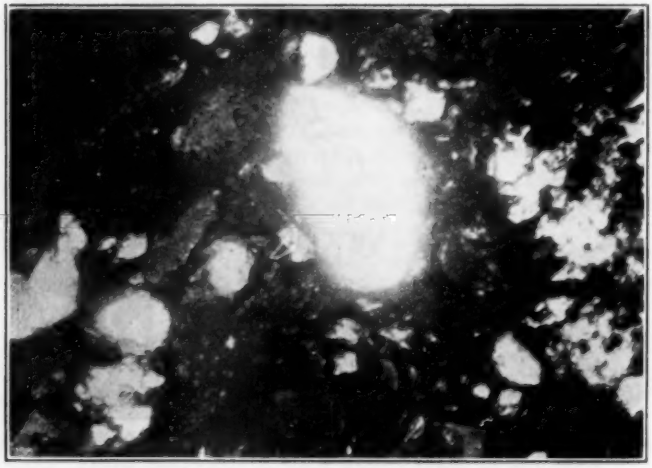

FIG. 35 BRICK NO. 35. UNUSED BRICK. MAGNIFICATION 210×. CROSSED NICOLS.

The same field shown in Fig. 34.

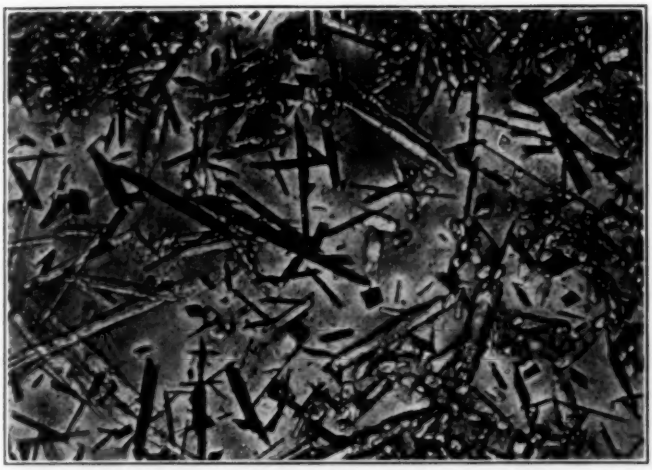

FIG. 36 BRICK NO. 22. SECTION OF SLAG FACE. MAGNIFICATION 210×. ORDINARY LIGHT.

Both primary and secondary mullite crystals in a matrix of brownish slag glass. Primary mullite, partly dissolved, in aggregates surrounding the field. Secondary mullite as long, well-developed needles associated with dark brown laths of prismatic crystalline material of unknown composition. Quartz absent, probably having been dissolved in the slag glass.

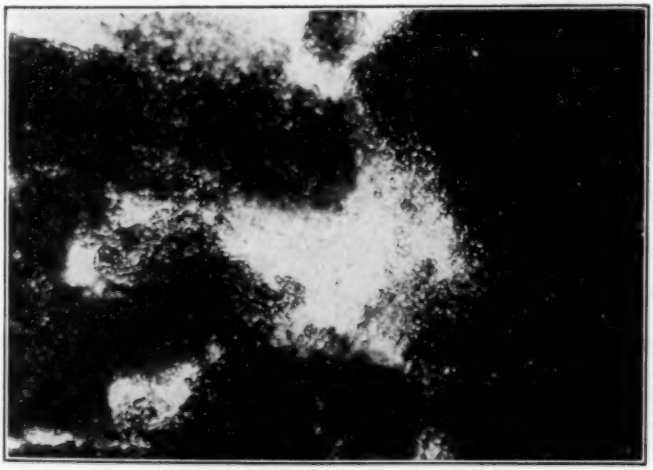

FIG. 37 BRICK NO. 22. SECTION ABOUT ¼ IN. FROM SLAG FACE. MAGNIFICATION 210 ×. ORDINARY LIGHT.

Aggregates of large, well developed primary mullite needles (about one-half the size of those in Fig. 36) embedded in slag glass. Mullite shows considerable solution. Secondary mullite and the unknown brown mineral of Fig. 36 are absent. Quartz grains absent.

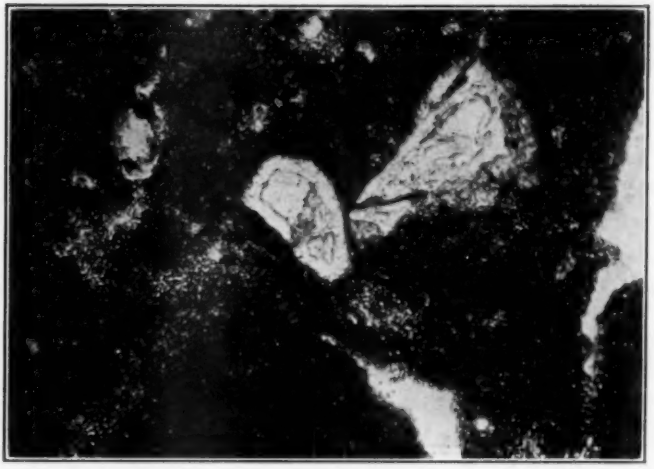

FIG. 38 BRICK NO. 22. SECTION ABOUT $\frac{5}{16}$ IN. FROM SLAG FACE. MAGNIFICATION 210 ×. ORDINARY LIGHT.

Aggregates of primary mullite crystals somewhat smaller than in Fig. 37. Glass somewhat less abundant and practically colorless. Small amount of quartz present, grains shattered and showing wide solution rims.

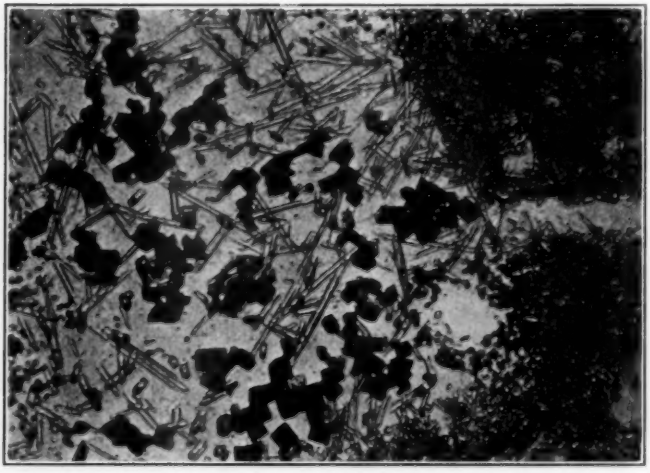

FIG. 39 BRICK NO. 48. SECTION IMMEDIATELY BACK OF SLAG FACE. MAGNIFICATION 210 ×. ORDINARY LIGHT.

Secondary mullite needles and dark brown octahedral and dodecahedral crystals of almandite garnet ($3FeO \cdot Al_2O_3 \cdot 3SiO_2$) in slag glass bordered by aggregates of partly dissolved primary mullite crystals.

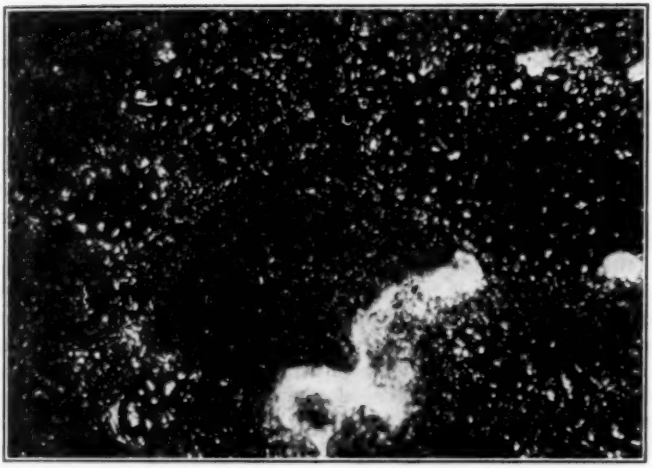

FIG. 40 BRICK NO. 65. SECTION ABOUT $\frac{1}{8}$ IN. BACK OF SLAG FACE. MAGNIFICATION 210 ×. ORDINARY LIGHT.

Mullite needles and glass surrounding nodular aggregates of rounded corundum grains. No quartz observed.

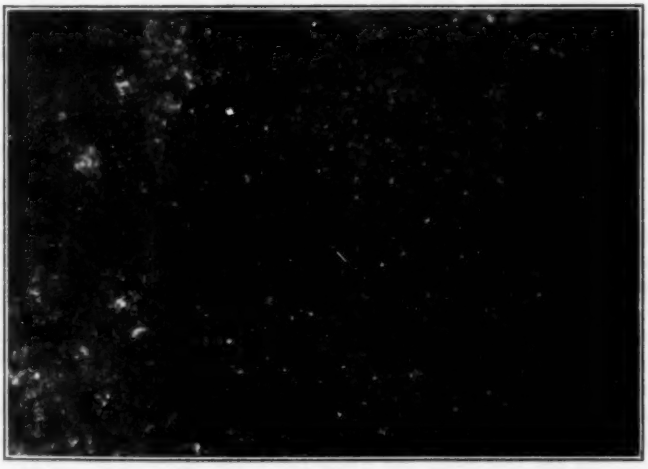

FIG. 41 BRICK NO. 65. SECTION $\frac{3}{8}$ IN. TO $\frac{1}{2}$ IN. BACK OF SLAG FACE. MAGNIFICATION 210 ×. ORDINARY LIGHT.

Minute mullite needles in glass surrounding large nodule of corundum.

The analysis of service conditions is the present function of the 1924 survey.

159 Fusion tests have been standardized which give thoroughly satisfactory determination of the limiting temperature to which the face of the brick may be exposed under most favorable conditions.

160 Tests have been devised and are in use for determining the deformation of brick under load at high temperatures. There is at present some diversity as to range of temperature and loads applied and these, it is believed, could well be placed on a basis of standardization.

161 Tests to determine the effect of slag upon refractory materials are now made by cone fusion of mixtures of slag and brick, and by various methods of applying molten slag to the refractory in the shape of comparatively large pieces or full-size brick. The cone type of test has a very valuable use in determining relative chemical activities and critical temperatures for brick and slag mixtures but cannot be accepted as the sole measure of slag action because of its failure to take account of the structure of the refractory as manufactured. This form of test could advantageously be standardized as to fineness and ratio of materials, and binder to be used, and as to composition of atmosphere to which the specimen is exposed.

162 There are two principal methods of tests in which the molten slag is applied to the uncrushed brick. Under one method,[22] pulverized slag in small amounts is fed at definite intervals into a trough-shaped recess previously prepared in the top of the brick which is set on end in a small reverberatory type of furnace and maintained at some predetermined temperature, depending upon the intended service conditions. It is important that the temperature of the test correspond approximately with the maximum temperature under which the brick are to be used, as comparatively small temperature differences may greatly affect the fluidity of the slag. This is the method of test employed in the case of the slagged specimens shown in Figs. 14, 15, and 16. Under the other method, small cylinders are molded or cut from the brick to be tested and set on the bottom of the test furnace immediately beneath the vertical nozzle of a high-pressure burner and, when the desired temperature is reached, pulverized slag is fed at definite uniform rate into the gas stream above the nozzle. As in the case of the first test, the time of application is fixed and the temperature is varied in accordance with probable service requirements. Definite control of atmosphere is of course essential under any method of test. In the test using the whole brick, the effect of slag is gaged by measuring depth of erosion and penetration. In the case of the other type of test, the excess slag is ground away from the specimen and a numerical value for slag resistance obtained from the relation of final and initial volume.

163 Both forms of test seem to possess considerable merit. The second test perhaps shows more nearly the effect of slag impingement; the test first described has the advantage of showing the cutting action of the slag on the unimpaired structure of the original brick, and very closely approximates conditions obtained in furnace side walls exposed to the flow of molten slag. It departs from simulation of service conditions to the extent that the entire brick, instead of merely an exposed face, is raised to the full furnace temperature and, for that reason, there is no increase of slag viscosity from cooling within the brick such as would normally occur. This condition, of course, accentuates the slag action and for laboratory test purposes is doubtless an advantage.

164 Tests have been developed to give indications of structural changes resulting from continued application of higher temperatures. The most important ordinarily used are those for determining constancy of volume and porosity. Such tests, as usually conducted, have been limited to determinations at 2552 deg. fahr. (1400 deg. cent.) which is satisfactory for most conditions of service but with increasing furnace temperatures it is desirable to apply supplemental tests at 2732 deg. fahr. (1500 deg. cent.). Tests of this type give indications of direct bearing upon the service life of the brick. They need careful interpretation, however, to avoid confusing with the effects of overfiring, slight expansions or contractions which may be of a secondary character. To these so-called " reheat " tests should be added petrographic examination, to determine the microstructure and changes which may be expected to occur, and X-ray examination which is now being applied to the study of the crystal structure in refractory materials and which, with further development, has great possibilities in this field.

165 Of all the commonly recognized physical properties of firebrick, resistance to spalling has been the most uncertain of determination. The usual spalling test consists of heating one end of the brick to a predetermined temperature and cooling more or less rapidly in water or air, the measure of resistance being the number of cooling cycles required to cause loss, at the heated end, of either the entire 2½ by 4½-in. face, or a definite percentage of the original weight. Various temperatures have been suggested for the water-cooled spalling test, the recommendation for a particular temperature being based chiefly upon facility of comparison afforded by the results. Only the maximum temperature suggested, however, is as high as that to which the brick ends are heated in service, and cooling in water simulates no condition normally met in the boiler furnace. The air-cooled spalling test in which the brick ends are heated slightly above the maximum which they may attain in the furnace and cooled by a high velocity air jet is also more severe than any thermal shock which the brick encounter in the ordinary furnace but is a far closer simulation of

service conditions. F. H. Norton has proposed, instead of the normal spalling test, determination of coefficient of expansion, diffusivity, and maximum shearing strain, and the calculation of spalling resistance by formula from these properties. The Mellon Institute Industrial Fellowship has developed a form of test in

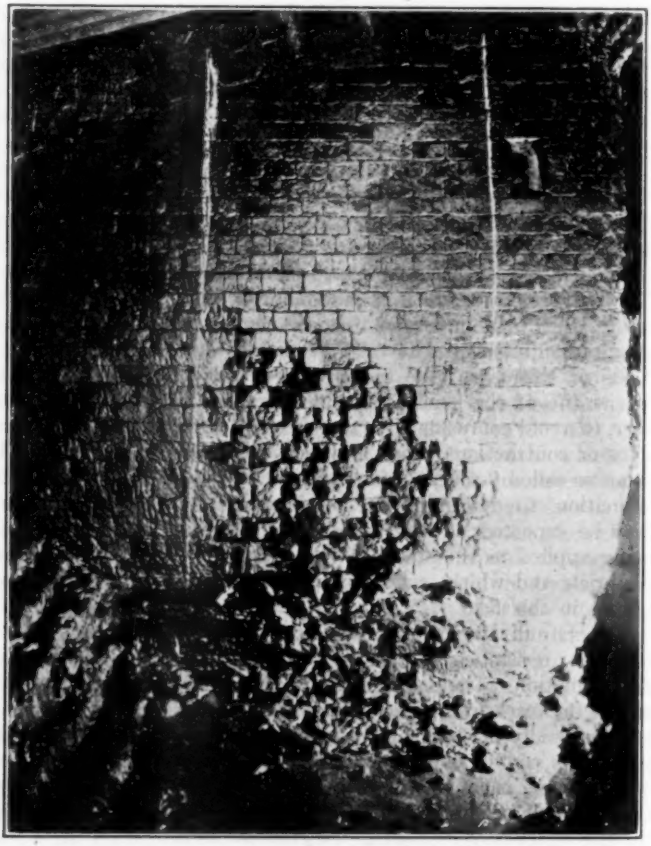

FIG. 42 SERVICE TRIALS, OIL-BURNING FURNACE

which a number of brick, set as in ordinary wall or arch construction, are subjected collectively to alternate heating and cooling under definite control. Fig. 42 shows the side wall of an oil-burning boiler furnace normally operated under very severe load fluctuations which is made use of in analyzing the spalling tendencies of refractories. None of these spalling tests, however, is able to give an indication of spalling resistance except as applying

to the brick in the condition tested; the value of results must in every case depend upon interpretation from other tests which give indication of susceptibility to structural change. A great deal more study can well be devoted to the problem of spalling.

166 For specification purposes, the Government very fittingly places tests under two classifications; eliminative or selective tests, and control tests. Selective tests may well cover any particular properties that the service conditions require such as fusion, load-carrying capacity, and spalling. With reasonable knowledge of the general properties of the refractory selected, control tests may frequently be simplified to the determination of a single variable, as volume changes on reheating at 2552 deg. fahr. (1400 deg. cent.) or at 2732 deg. fahr. (1500 deg. cent.) for example.

167 Specifications for refractories which include test requirements should conform as closely as practicable to the actual needs of the service to which they will be applied. There is no advantage to be gained from a specification of high resistance to slag in the average anthracite-burning furnace, for example, when this quality is had at the sacrifice of resistance to spalling. Nor is extremely high refractoriness always desirable. If there is no softening of the heated end of the brick, unless well protected by slag, the majority of refractory materials will fail from expansion stresses.

168 A wider recognition of characteristics inherent in the nature of refractories will avoid many of the difficulties at present experienced by users. A wider recognition, on the part of manufacturers, of the possibilities of their raw materials will bring a general raising of production standards and will enable them better to meet the continually advancing demands of the modern boiler furnace.

SPECIAL REFRACTORIES

169 A number of special refractory materials have been suggested for boiler-furnace use, as measures of combating clinker adhesion and the fluxing and erosive action of slag. Recently, however, such materials have been suggested to provide greater refractoriness to meet increasing furnace temperatures. The essential composition and some of the characteristics of the more important of these special materials are given in Table 8.

Silicon Carbide

170 As applying strictly to boiler-furnace construction, silicon carbide has been the most widely used of the special refractory materials and, until very recently, it was the only such material which could be regarded as commercial.

171 As is well known, silicon carbide is an electric-furnace product [25] resulting from the reaction between coke and silica sand at a temperature of about 3344 deg. fahr. (1840 deg. cent.). In the pure state it is not subject to fusion, but at about 4064 deg. fahr. (2240 deg. cent.) silicon carbide is decomposed. It maintains

TABLE 8. PHYSICAL PROPERTIES OF REFRACTORIES

		Silicon carbide							
	Firebrick	Recrystallized	Bonded	Fused Alumina (bonded)	Bauxite	Kaolin	Magnesia	Chrome	Zirconia
Composition	15-45% Al_2O_3 40-75% SiO_2	90-95% SiC	85-90% SiC	89% Al_2O_3	55-95% Al_2O_3	96% Al_2O_3. $2SiO_2$	80-94% MgO	35-48% Cr_2O_3	75-95% ZrO_2
Fusion point, deg. cent.	1500-1750	Decomposes	Decomposes	1750-2000	1720-1840	1755	2150-2165	1850-2050	Above 2000
deg. fahr.	2732-3182	.....	.....	3182-3632	3128-3344	3191	3902-3929	3662-3722	Above 3632
Porosity (per cent)	10-30	20-36	14-20	15-20	30-40	22-27	22-35	15-28	19-25
Bulk specific gravity	1.8-2.2	2.0-2.5	2.4-2.6	3.0	1.9-2.0	1.9-2.1	2.30-2.75	3.0-3.2	3.6-4.0
Specific gravity	2.2-2.8	3.1	3.1	3.8	3.1-3.3	2.6	3.4-3.6	3.8-4.5	4.8-5.0
Mean specific heat									
25- 100° C. 77- 212° F.	0.20	0.18	0.19	0.19	.....	.....	0.23	0.17	.....
25-1000° C. 77-1832° F.	0.26	.....	.....	.....	.....	.....	0.28	0.20	0.16
25-1500° C. 77-2732° F.	0.30	.....	.....	.....	.....	.....	0.30	0.22	0.18
Mean thermal conductivity									
Gm.Cal.Sec.$^{-1}$Deg.C.$^{-1}$Cm.$^{-2}$Cm.									
25-1000° C.	0.0025-0.0035	0.024	0.0205	0.0079	0.002-0.004	0.0046	0.010-0.016	.....	.....
B.t.u.Hr.$^{-1}$Deg.F.$^{-1}$Ft.$^{-2}$In.									
77-1832° F.	7.25-10.15	69.6	59.4	22.9	5.8-11.6	13.33	29.0-46.4	.....	.....
Mean coefficient of expansion									
Per deg. cent.	0.000006	0.0000045	0.0000045	0.0000068	0.0000053	0.0000043	0.000014	0.000011	0.000006
Per deg. fahr.	0.000003	0.0000025	0.0000025	0.0000038	0.0000029	0.0000024	0.000008	0.000006	0.000003

See reference 38.

practically full crushing strength to 2462 deg. fahr. (1350 deg. cent.), and at temperatures as high as 3092 deg. fahr. (1700 deg. cent.) shows no evidence of softening under a compressive load of 175 lb. per sq. in. The material is furnished for commercial use both in the practically pure form and bonded with refractory clay. In the latter form the brick are usually fired in periodic kilns at temperatures around 2462 to 2642 deg. fahr. (1350 to 1450 deg. cent.).

172 On account of their mechanical strength, infusibility, and high thermal conductivity, silicon-carbide refractories under sufficiently low temperature conditions have been found very effective in resisting certain kinds of slag adhesion. In a strongly reducing atmosphere silicon carbide is practically unaffected by slags. On the other hand, where air is present oxidation begins at temperatures as low as 1550 deg. fahr. (843 deg. cent.), causing gradual growth or permanent expansion due to conversion of SiC to SiO_2. At around 2200 to 2500 deg. fahr. (1204 to 1371 deg.

cent.) a protective glaze* consisting largely of silica, iron, and alumina, forms, which very materially slows down the rate of oxidation, but at higher temperatures the glaze becomes too fluid to be effective. Also, in an oxidizing atmosphere silicon carbide is decomposed by slags of high iron content at ordinary furnace temperatures, forming ferrosilicates with very destructive effects. As an illustration, the Illinois coal slag of Table 6, prepared in test-cone form [22] — ratio 20 per cent slag with 80 per cent silicon carbide — showed softening as low as 2650 deg. fahr. (1454 deg. cent.) with complete fusion and entire decomposition of silicon-carbide

TABLE 9 ANALYSIS OF SPECIAL REFRACTORY MATERIALS

	California Andalusite	Grecian Magnesite	Spinel (brick)	Chromite
Ignition	3.67	5.48	....	1.81
Silica	33.78	3.84	1.00	4.44
Alumina	56.89	....	65.30	21.65
Iron oxide	5.37	0.65	0.60	17.28
Titania		....	....	....
Lime		3.46	0.70	trace
Magnesia		83.23	36.85	15.35
Alkalis		....	....	....
Carbon dioxide	....	3.34	....	....
Chromic oxide	....	....	....	40.08

	No. Carolina Cyanite	Missouri Diaspore	Arkansas Diaspore	Arkansas Bauxite
Ignition	0.20	13.67	14.36	28.08
Silica	36.68	7.20	4.40	9.70
Alumina	59.87	73.60	76.28	56.31
Iron oxide	trace	1.78	1.46	3.10
Titania	0.40	3.27	2.52	2.09
Lime	0.56	0.13	0.39	0.22
Magnesia	0.56	0.22	0.43	0.52
Alkalis	....	0.43	1.00	0.49

	Georgia Kaolin		Guiana Bauxite	
	Clay	Brick	Clay	Brick
Ignition	12.62	....	33.00	....
Silica	45.49	51.94	1.70	32.46
Alumina	39.96	45.62	65.00	63.94
Iron oxide	0.30	0.95	0.10	1.14
Titania	1.64	1.49	0.90	2.30
Lime	....	0.13	0.00	trace
Magnesia	....	0.12	....	0.38
Alkalis	....	0.41	....	....

crystals at 2760 deg. fahr. (1516 deg. cent.) in the standard fusion test. At lower rates of heating, softening and fusion would have occurred at much lower temperatures. For this reason silicon-carbide refractories must be artificially cooled to develop their full possibilities in the boiler furnace.

Alumina-Silica Materials

173 The special refractories of next importance at the present time belong to the alumina-silica group: the highly aluminous materials in which silica occurs only as an impurity, diaspore,

* Manufacturers now make a practice of applying such a glaze in finishing the brick; however, it is not always effective.

bauxite, and fused alumina; kaolin; the sillimanite group of materials, sillimanite, andalusite, and cyanite; and the synthetic product, mullite.

174 Bauxite was the first of these materials to be given commercial trial, but the shrinkage characteristics of highly aluminous materials were not then fully understood and such great difficulty was encountered from continued shrinkage in service that its use in boiler furnaces was soon discontinued. Diaspore, which is also subject to high shrinkage but not quite to the same degree as bauxite, has been used in refractories within the past few years and at the present time, although few manufacturers seem to appre-

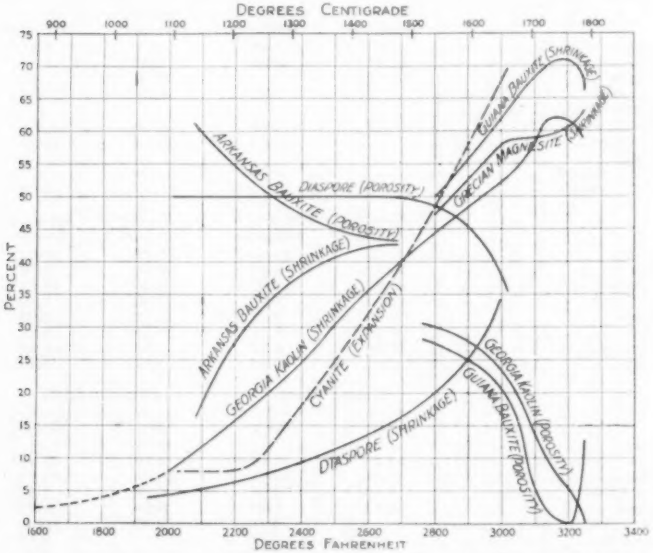

FIG. 43 BURNING CHARACTERISTICS OF SPECIAL REFRACTORIES

ciate the requirements of shrinkage removal, the diaspore refractory may be referred to as a commercial product.

175 Bauxite and diaspore are found in commercially workable deposits in this country chiefly in Arkansas and Missouri. Somewhat similar material is found in Georgia, and bauxite is now being imported for refractory purposes from Dutch Guiana. Analyses of the native and Guiana materials are given in Table 9. The chemical formula for diaspore is $Al_2O_3.H_2O$ and bauxite is considered to be a mixture of $Al_2O_3.H_2O$ and $Al_2O_3.3H_2O$ or gibbsite. Shrinkage and porosity characteristics of Missouri diaspore and for the high-temperature range of the Guiana bauxite are shown in. Fig. 43. It will be noted that the shrinkage of the bauxite which has reached 50 per cent at 2800 deg. fahr. (1538 deg. cent.), con-

tinues at a high rate to 3200 deg. fahr. (1760 deg. cent.) the point of overfiring. It is this characteristic which was responsible for the failure of the early bauxite refractories which were burned in commercial kilns with ordinary fireclay brick. The diaspore shrinkage is not excessive below 2800 deg. fahr. (1538 deg. cent.) but at about that temperature begins increasing at a very high rate. Other specimens of diaspore burned at a temperature of 2966 deg. fahr. (1630 deg. cent.) have shown a shrinkage of about 20 per cent. Experience with these materials, however, has indicated that shrinkage is affected by time of heating as well as by temperature attained. The definite relations of time and temperature in producing shrinkage are not yet determined.

176 The diaspore refractories are generally mixtures of diaspore and fireclay. On account of their higher alumina content they have higher fusion points and, when properly burned, greater resistance to deformation under load at higher temperatures than fireclay brick. They have a high resistance to certain of the iron slags but, on account of the low melting point of the eutectics formed, are fluxed by some coal slags, especially where containing a high percentage of lime. The brick shown as specimen No. 12 in Fig. 14 is of diaspore composition. On the other hand, as in the case of fireclay brick, the slag resistance is doubtless affected very markedly by texture and burn. Some of the diaspore refractories show very high spalling resistance but this property is also dependent upon texture and burn as well as composition.

177 One of the manufacturers is now developing a brick made largely of Guiana bauxite. The analysis of this brick as now made — it is still at the laboratory stage — is given in Table 9. The composition is very close to that of mullite in alumina-silica ratio. Under the 25-lb. load test the brick gives the first evidence of softening at 2750 deg. fahr. (1510 deg. cent.) as compared with about 1800 to 2200 deg. fahr. (982 to 1204 deg. cent.) for the usual first-quality firebrick. Its spalling resistance is higher than that of the usual first-quality firebrick. The slag-resisting characteristics are not yet fully determined but will, of course, necessarily be limited by the eutectics of the alumina-silica minerals. The erosion of Guiana bauxite in the slag-impingement test, in comparison with that of other refractory materials, is shown in Fig. 44. The slag used in the Illinois coal slag referred to in Table 6.

178 Electrically fused alumina is commonly made from Arkansas bauxite and is, of course, entirely free from shrinkage in service. On account of its greater density it is also more resistant to the action of slags than the usual kiln-burned diaspore refractories but, like other aluminous materials, it is attacked by lime, as is to be expected from the equilibrium diagram of Fig. 2. Refractories made of fused alumina have usually been bonded with fireclays. This bond has necessarily limited the refractoriness of the

product. At the present time, however, a highly aluminous bond is being developed and it is expected that the refractoriness of the product will closely approach that of the pure fused alumina which remains practically rigid close to its fusion point of 3722 deg. fahr. (2050 deg. cent.).

179 Deposits of kaolin of exceptional purity are found in parts of the southeast, but owing to their continued shrinkage they have only recently been successfully employed in firebrick for boiler-furnace purposes. Analysis of kaolin from one of the Georgia deposits is shown in Table 9, and the shrinkage and porosity characteristics of this material as fired to its temperature of fusion are shown in Fig. 43. Refractory brick are now being made of this material by calcining at 3000 deg. fahr. (1649 deg. cent.) all except a very small percentage of the clay which is required to serve as

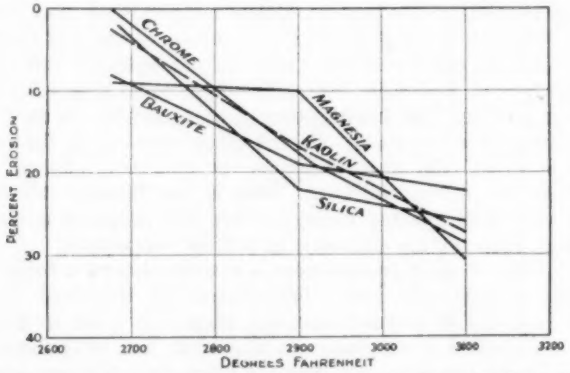

FIG. 44 SLAG TESTS OF SPECIAL REFRACTORIES: ILLINOIS-COAL SLAG

bond. The molded brick is also fired at approximately 3000 deg. fahr. (1649 deg. cent.) the final texture is shown in Fig. 26. High-temperature load-carrying characteristics of the kaolin brick do not show quite as high refractoriness as the Guiana bauxite but softening under 25-lb. load begins only at 2650 deg. fahr. (1454 deg. cent.). In some instances under high-temperature conditions it has been remarkably free from spalling. The lower impurity content and more completely crystalline structure of this brick give it greater resistance to some slags than is had from the normally burned fireclay brick of corresponding composition.

SILLIMANITE, ANDALUSITE, AND CYANITE

180 Sillimanite is found in this country and, so far as known, on this continent only in small quantities and in no instance in commercial amounts. Andalusite is now mined in California and is reported in Idaho and Wyoming. Cyanite is found in large

deposits in North Dakota, Wyoming, California, and Virginia. The refractory possibilities of these minerals are now under experimental investigation by the Bureau of Standards, Bureau of Mines, and by some commercial manufacturers.

181 These three minerals, which are of identical chemical composition but of different crystallographic formation, are decomposed by heat to mullite and silica. Cyanite is the most readily decomposed, the change beginning very slowly about 2012 deg. fahr. (1100 deg. cent.) and taking place fairly rapidly around 2498 deg. fahr. (1370 deg. cent.). Andalusite seems to require somewhat higher temperature, about 2534 deg. fahr. (1390 deg. cent.), for decomposition. Under prolonged heating the change might possibly begin at lower temperature and it is probable that the rate of change is affected by the impurity content. Sillimanite is not decomposed on heating below 2813 deg. fahr. (1545 deg. cent.) but at that temperature is converted to mullite and high-silica glass.

182 The decomposition of these minerals is necessarily accompanied by increase of volume. As quoted by J. W. Greig,[37] the specific gravities of cyanite, andalusite, and sillimanite according to Dana[7] are 3.56-3.67, 3.16-3.20, 3.23-3.24, respectively, whereas the specific gravity of mullite according to Greig is 3.156 and that of the resulting "free" silica will vary between 2.333 and 2.210. A. B. Peck[38] gives the following changes of specific gravity in firing the materials to cone 15, approximately 2606 deg. fahr. (1430 deg. cent.):

	Unfired	Fired to cone 15
Andalusite	3.29	3.20
Cyanite	3.59	3.09
Sillimanite	3.15	2.92

Cyanite shows the highest change of volume and andalusite the least change, as the result of firing to this temperature.

183 Andalusite, according to Peck, shows only slight change in volume at its dissociation point, and practically no disintegration of structure. This material is being employed experimentally in the manufacture of refractories without previous calcining. Preliminary trials promise the development of a high-grade refractory; actual experience has not been sufficient for definite determination of the ultimate characteristics.

184 F. H. Riddle in his experimental work on andalusite brick, which were of slightly lower than 1:1 alumina-silica ratio and included about 5 per cent of fluxes, has measured the shrinkage occurring during a year's exposure in a porcelain kiln subjected to temperatures up to about 2700 deg. fahr. (1482 deg. cent.). Brick of fine-grained structure, made of 100-mesh powder, showed a volumetric shrinkage of about 45 per cent; relatively coarse-grained brick, made up of material screened through 8 mesh, on

the other hand, showed a shrinkage of not more than 8 per cent. Evidently, as suggested by Riddle, in the large grains shrinkage occurred by increase of absolute density and increase of apparent porosity, but in the 100-mesh powder the structure was so far destroyed that the individual particle could no longer act as a separate unit, and mass shrinkage occurred. This experience is a remarkable illustration of the effect of screen size on shrinkage and seems to apply, at least in a general way, to most refractory raw materials.

185 In his laboratory tests of andalusite brick Riddle found that those made exclusively of andalusite failed rapidly in the usual type of water spalling test. Petrographic studies showed that crystal growth did not occur to any great extent between grains where andalusite alone was used, but did occur with a clay bond. This experience suggested that the spalling tendency first shown by the andalusite brick might be overcome by bonding with clay, and this plan is being followed.

186 The expansion of cyanite in decomposing is so great and apparently occurs with such violence that, as pointed out by Greig,[34] actual rupture of crystal structure occurs. On account of this expansion it was suggested by Peck[35] that cyanite could not be advantageously used in the production of refractories without previous calcining. G. A. Bole and H. C. Harrison of the Bureau of Mines, however, have taken advantage of the expansion of cyanite to combine it with diaspore without calcining either material, and find it possible to produce a mixture having an alumina-silica ratio somewhat below that of mullite which can be fired to cone 26, about 2912 deg. fahr. (1600 deg. cent.) and be identically die-sized. Brick made of these materials with alumina and silica in the exact mullite ratio, or with a slight excess of alumina, show some shrinkage at that temperature. Bole has obtained the following shrinkage-porosity characteristics for mixtures of equal parts of cyanite and diaspore, using cyanite of rather coarse grind in mixture A and finer grind in mixture B; whether prolonged heating, as under actual service conditions, will change these values is still to be determined.

Burn	Cone	14	20	26	30
	Approximate Temperature	2570°F. 1410°C.	2786°F. 1530°C.	2912°F. 1600°C.	3038°F. 1670°C.
Mixture A:		(Per cent)	(Per cent)	(Per cent)	(Per cent)
Linear expansion (over die-size)		3.5	3.5	3.5	2
Porosity		37.4	35.8	34.8	34.1
Mixture B:					
Linear expansion (over die-size)		1	0	−1	−4
Porosity		37.1	32.6	32.9	26.2

The expansion characteristics of cyanite and the shrinkage of diaspore are shown in Fig. 43.

187 The cyanite brick is also at the laboratory stage. However, one of these, a cyanite-clay brick, installed experimentally in a suspended arch subject to rapid deterioration from spalling and slag erosion, has shown no evidence of spalling and a higher resistance to slag than the high-grade clay brick customarily used.

Other Alumina-Silica Brick

188 Brick of various alumina-silica ratios have been made experimentally by Wilson, Sims, and Schroeder[36] using electrically fused materials with clay bond. These investigators found that in mixtures having less than the mullite ratio of alumina and silica the structure was very much weaker than when mixtures of mullite or higher ratio were used. Apparently in mixtures where the alumina was 68 per cent or less, even though the brick were fired to cone 33, 3128 deg. fahr. (1720 deg. cent.) over a period of twenty-four hours, insufficient recrystallization occurred to give proper strength of structure, or else, as has been otherwise suggested, the crystallization of mullite under these conditions causes the concentration of impurities in the uncombined silica and, as this silica serves as a cement between mullite crystals, in lowering the softening point of the silica the entire mass is correspondingly affected. None of the test brick made from these lower-ratio mixtures were able to resist the 25-lb. load test above cone 18, 2715 deg. fahr. (1491 deg. cent.). However, when the alumina was raised to 70 per cent or higher, an interlocking structure of corundum and mullite was obtained, and brick fired to cone 33, 3128 deg. fahr. (1720 deg. cent.), cracked, but showed no compressive deformation under 25 lb. load at cone 32, 3101 deg. fahr. (1705 deg. cent.). Another brick, fired to cone 18-20, 2750 deg. fahr. (1510 deg. cent.), over a period of forty-eight hours, was entirely unaffected in the 25-lb. load test at cone 29, 3002 deg. fahr. (1650 deg. cent.).

189 In comparing the slag-resisting qualities of these higher-alumina brick with those of other refractories Wilson, Sims, and Schroeder[36] found that at temperatures above 2822 deg. fahr. (1550 deg. cent.) the "sillimanite-corundum" or mullite brick had high resistance to acid slags and, while attacked by basic slags, showed higher resistance to open-hearth slags of this character than either silica or magnesia (the magnesia brick, however, going to pieces chiefly by spalling), and higher resistance to iron-oxide slag than zirconia or silicon carbide. These investigators suggest that it is on account of their greater density that the artificially prepared brick have shown this resistance to slag penetration. For the same reason, however, the resistance to spalling has been comparatively low.

190 Alumina-silica refractories of any composition or method of manufacture cannot resist the fluxing action of slags at temperatures materially beyond the melting points of the corresponding

eutectics. However, the purer materials tend to form eutectics of higher melting points. Also, as pointed out by Wilson, Sims, and Schroeder, crystalline materials have the ability to resist deformation at temperatures very close to their melting points, and, through the greater density and greater rigidity of their crystalline structure, they may be expected to have somewhat higher slag resistance than brick of corresponding chemical composition made from clays as ordinarily burned.

Magnesia, Spinel, Chrome, Zirconia, and Zircon

191 A number of other special materials are being used or are under development for refractory purposes. The more important of these are magnesia, spinel, chrome, zirconia, and zircon. All of these materials, with the possible exception of zircon, require very high temperature calcining or fusion in the electric furnace to avoid continued shrinkage in service under boiler-furnace conditions. In general, brick made from these materials have been bonded with a relatively low-temperature flux or with clay, and in consequence have not shown satisfactorily high-temperature load-carrying capacity, and, also partly on this account, have had very poor resistance to spalling. Wilson, Sims, and Schroeder [36] have suggested that it should be possible, by the use of sufficiently high temperatures, to develop an interlocking crystalline structure in these materials as has been done in the case of higher-grade silicon-carbide brick. They point out that when clay or amorphous material is used for bonding purposes the softening of the brick will necessarily vary directly with the amount and character of the amorphous bond.

192 Magnesia, or as more commonly known, magnesite, has been largely employed in steel furnaces in the calcined or "dead-burned" state. The brick are made from magnesite ore which previous to the war was almost exclusively imported from Europe. There are, however, large magnesite deposits in the states of Washington and California, and these have since been developed. For refractory purposes magnesite must be calcined at high temperature to expel the CO_2 and to sinter the particles together; for complete avoidance of shrinkage in service the magnesia should be converted to the crystalline form of periclase. The temperature required to effect complete shrinkage will vary with the texture and composition of the crude magnesite from 2642 to 3092 deg. fahr. (1450 to 1700 deg. cent.), condensation of structure being facilitated by the presence of certain impurities, as iron oxide. The specific gravity of the amorphous magnesia is about 3.10 and that of periclase is 3.67. The magnesite curve of Fig. 43 gives some indication of the burning shrinkage.

193 The greater part of the available data on magnesia refractories has been obtained from the ordinary kiln-burned materials

which will in general contain from 4½ to 8 per cent ferric oxide. The ferric oxide is present in some grades of magnesite ore in sufficient quantity for sintering purposes, but in others it is added. The magnesia refractories also will rarely be completely converted to periclase. The specific gravity of the brick will in consequence vary from about 3.40 to about 3.60. The data given in Table 8 refer to the commercial kiln-burned brick.

194 Magnesia as a refractory material has advantages of high fusion point and resistance to basic slags, but on the other hand its eutectic with silica has a melting point[37] around 2809 deg. fahr. (1543 deg. cent.) and in the presence of impurities this melting point falls much lower. The high rate of expansion combined with relatively weak bond of the commercial brick has made the magnesia refractories very susceptible to spalling. At temperatures around 2642 deg. fahr. (1450 deg. cent.) they have been found subject to very slow corrosion by carbon, the deterioration increasing with rising temperature and proceeding fairly rapidly around 3632 deg. fahr. (2000 deg. cent.). The kiln-burned magnesia refractories are also subject to slaking on exposure to moisture. On account of the weakness of bond they cannot be safely used under load-test conditions at temperatures much above 2552 to 2732 deg. fahr. (1400 to 1500 deg. cent.). Some investigators,[38] however, have found that the strength of magnesia is greatly increased by the addition of alumina, chromium oxide, titania, or zirconia. It is possible that, as in the case of the alumina-silica refractories, superior materials for some requirements can advantageously be derived from combinations of these more special materials.

195 Electrically fused magnesia of low iron content is now produced by some manufacturers. This material is not affected by atmospheric moisture and is free from further volume change. These characteristics are, of course, of marked advantage over the comparative instability of the ordinary calcined material. In the molded brick, however, the strength of the structure is dependent upon that of the bond used and, as suggested by C. E. Williams,[33] the development of a satisfactory magnesia refractory for boiler-furnace construction may possibly require the use of an electrically sintered bond.

196 The eroding effect upon magnesia brick of the Illinois slag used in tests of fireclay brick is indicated in Fig. 44, showing the results of impingement tests at 2700 deg. fahr. (1482 deg. cent.), 2900 deg. fahr. (1593 deg. cent.), and 3100 deg. fahr. (1704 deg. cent.). The magnesite used in the brick tested was of an exceptionally pure grade of the Grecian ore, the analysis of which is given in Table 9.

197 Spinel is an electric-furnace compound of alumina and magnesia of the formula $MgO.Al_2O_3$. The melting point[39] of the pure material is approximately 3875 deg. fahr. (2135 deg. cent.). Although both magnesia and alumina are considered basic refrac-

tories, the compound spinel is reported to be neutral and to have high resistance to slag. On account of the low melting point of the magnesia-alumina-silica eutectic, however, it would be surprising if spinel were able to resist the fluxing action of certain of the high-silica coal-ash slags. The electrically fused spinel has about the same coefficient of expansion as ordinary fireclay brick, and on that account it would seem that properly bonded spinel refractories should have a high resistance to spalling. The commercial materials, however, have been sensitive to temperature change and under 25 lb. load have failed abruptly at about 2400 to 2500 deg. fahr. (1316 to 1371 deg. cent.) due to the softening of the bond. Spinel refractories have not so far been commercially applied to boiler-furnace construction.

198 Chrome refractories are made from chromite, a double oxide of iron and chromium. More correctly, the chromium ore is a mixture of various minerals of the spinel group and is of widely varying composition, chromite, however, being the dominant mineral. The other mineral constituents, besides Cr_2O_3 and FeO, will commonly average about MgO, 15 per cent; Al_2O_3, 12 to 22 per cent; SiO_2, 4 to 8 per cent. The ore is found in several parts of the continent, notably in Oregon and California but, on account of the expense of development and variable quality of the domestic material, practically all of the commercial chrome refractories are made from imported ore. The material has been widely employed in the steel industry and is now being introduced in some of the special cements for boiler-furnace refractories. The commercial chrome brick is made from chromite containing about 35 to 48 per cent chromic oxide and is generally molded either by power press or by hand without other bonding material. As in the case of the magnesia brick, the commercial chrome refractory has been very susceptible to spalling due to high coefficient of expansion and weakness of bond; it is also subject to reduction at high temperatures in the presence of carbon. The melting point of the purer material[38] is 3956 deg. fahr. (2180 deg. cent.), and that of the commercial brick may be as high as 3722 deg. fahr. (2050 deg. cent.). In the load test,[39] the chrome brick fail suddenly at about 2642 deg. fahr. (1450 deg. cent.) under a 50-lb. load. In the steel industry chrome refractories are found to be highly resistant to both basic and acid slags. The erosive effect of the Illinois slag, as indicated by the impingement test upon a particular grade of chromite of analysis given in Table 9, is shown in Fig. 44.

199 Zirconia is an ore of formula ZrO_2 found chiefly in Brazil. In the pure state it has a melting point of 4928 deg. fahr. (2720 deg. cent.) and is reported[33] to be basic in character and highly resistant to slags. For refractory use the raw material must be thoroughly purified by chemical treatment and calcined at high temperature. The refined and calcined zirconia is highly porous,

devoid of plasticity, and has been found difficult to bond. When heated in air, zirconia tends to form nitrides, and when heated in the presence of carbon, it tends to form carbides. Zirconia has been reported to have an extremely low coefficient of expansion [3] and should therefore be very insensitive to sudden temperature change. On the other hand, some of the commercial brick have a fairly high coefficient of expansion and, in tests made at the University of Washington,[36] the commercial zirconia brick failed rapidly under the spalling test. Subjected to iron-oxide slag, they disintegrated after about one-half hour at about 2800 deg. fahr. (1538 deg. cent.). It is possible, however, that the electrically fused products of this material may show superior refractory properties.

200 Zircon, $ZrSiO_4$ is a compound of zirconia and silica which has been recently found in large deposits in the southeast, chiefly in Florida. Its physical properties are not yet definitely known. The reports of preliminary investigations of zircon were very promising. They indicated that the raw material would require no chemical treatment or other special purification before use and would undergo no shrinkage when heated to 3182 deg. fahr. (1750 deg. cent.) and that the molded refractory would not only be free from the tendency found in zirconia to form carbide in the presence of CO but be unaffected by slag. In the attempts that have been made to utilize zircon commercially, however, great difficulty has been encountered in securing a satisfactory bond and some shrinkage has been observed at temperatures of 2732 deg. fahr. (1500 deg. cent.) and above; it also appears that, in the commercial degree of purity, zircon is not altogether resistant to slag. Just what could be effected by further purification and by calcining at high temperature or by electrical fusion has not been determined. The pure zircon is unquestionably very highly refractory. The melting point [23] is given as approximately 4622 deg. fahr. (2550 deg. cent.). The thermal conductivity and the coefficient of expansion are understood to be very low as in the case of zirconia. Apparently the material should have valuable refractory possibilities.

201 All of the special refractory materials so far suggested, with the possible exception of zircon and zirconia, are subject to deterioration under conditions to which ordinary boiler-furnace refractories are now exposed. None of them, except zircon and zirconia, show higher melting points for eutectics possible in the boiler furnace than do refractories which can be made from alumina and silica. The deposits of zircon in Florida would seem to promise the possible commercial development of refractories of this material, but, excepting that one prospect, if furnace temperatures are to be carried beyond those which can be sustained by the alumina-silica group of refractories, artificial cooling will be required.

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DISCUSSION

H. G. Thielscher.[1] The selection of proper refractories is a difficult problem and the paper under consideration points out, in a most commendable manner, some of the factors to be considered and some of the pitfalls to be avoided. In the modern central station with high-set stokers or large furnaces for powdered fuel, the furnace cost has become of major importance and oftentimes is at least half the cost of the boiler.

Boilers as now operated with distilled water and provided with furnaces of sufficient size to complete combustion in the furnace are particularly free of trouble and the need for shutdown on account of boiler troubles has been reduced to a very small percentage. Analysis of the operating records of 12 large high-pressure boilers representing a period of 165,000 hr. shows the average total outage hours due to various causes to be 21 per cent of the hours in the period. Of this, the outage hours due to brickwork repairs were twice those due to all other causes. In the case of one boiler, representing a period of about nine months, the outage due to brickwork repairs has been reduced to zero. The brickwork repairs were confined principally to sections of the side walls where there is flame impingement and erosion. This erosion is aggravated by the high iron content and low fusing point of the ash. Extended trials of at least 20 different refractories used in the eroded areas failed to bring out one of noticeable superiority. These figures apply only to pulverized-coal firing with low-grade fuel running about 10,000 B.t.u., 2000 deg. fahr. fusing point of ash, 5 to 6 per cent sulphur, and 18.5 per cent iron in the ash. One of the 12 boilers is equipped with water-cooled side and back walls and air-cooled front walls. The other boilers have various combinations of air- and water-cooled refractories. The reduction possible in outage hours is to be noted in the case where during a period of nine months no brickwork repairs have been necessary or appear to be needed in the near future. This shows the benefits accruing from improvements in furnace design and operation.

[1] McClellan & Junkersfeld, Inc., New York, N. Y. Assoc. Mem. A.S.M.E.

The boiler horsepower installed per kilowatt of generating capacity depends primarily on the maximum rating which can be efficiently developed and on the boiler outage hours, and these depend on the furnace. We must not forget when comparing station designs to consider that the plant with low-grade coal is confronted with a much more difficult refractory problem than in the case where the ash fuses at 2500 deg. fahr. and has a sulphur content of 1 per cent or less.

One boiler of about 2000 hp., complete with furnace, firing equipment, draft facilities, building structure and other accessories, may represent an investment of a quarter of a million dollars. A reduction in this cost may be effected by increasing the reliability of furnaces and thereby reducing the number of spare boilers. A further saving may be realized in fuel and first costs by constructing furnaces to withstand higher temperatures and capacities. It is to be expected that the work now being undertaken by this Society and by the American Institute of Refractories will result in further benefits to the users and more clearly define the limitations of various refractories, particularly with reference to service conditions.

L. E. HANKISON.[1] During the years 1924 and 1925 the West Penn Power Company conducted tests on 30 brands of firebrick and 23 brands of refractory bonding material at the Mellon Institute of Industrial Research, Pittsburgh, with an idea of finding either a material with which to patch eroded walls or a material for new walls that would withstand the ravages of slag erosion. As an example of the erosion existing under very bad slag conditions, Fig. 45 shows a back wall cut by slag in one of the boiler furnaces at our Windsor Power Station.

In order thoroughly to investigate the problem, it was necessary to reproduce on a laboratory scale the conditions found in our boiler furnaces. After much experimenting with the design of furnaces, one was developed in which four different specimens of firebrick could be compared at one test under the same conditions of temperature, pressure, duration of run, and rate of slag feed. Fig. 46 shows the furnace after the roof had been removed, but shows the test specimens still in place under a spring loading of 1000 lb. or 25 lb. per sq. in., which was maintained throughout each test.

Fig. 47 shows a sectional elevation of the furnace with the test walls in place under pressure and the arch roof sprung against these test walls. Four slag ovens are built on the roof and the gases of combustion pass over the slag, melting it and causing it to drip on to the test walls, producing erosion similar to that shown in Fig. 45. The test specimens used in this furnace are 9 in. wide

[1] Supt. of Efficiency Dept., West Penn Power Co., Pittsburgh, Pa. Mem. A.S.M.E.

and 15 brick high with joints in the center of every other brick, so as to reproduce as nearly as possible the conditions in a boiler where slag flows over the joints of the wall. This tests the mortar with which the brick are laid as well as the brick, and really reproduces boiler-furnace walls except that the test wall is only ten inches thick. Natural gas supplied through four burner pipes, one in each corner pillar, is the heating medium, the flames impinging upon each other in a downward direction and then coming up the walls and out the roof over the slag ovens. The molten

FIG. 45 ERODED BACK WALL AT WINDSOR STATION

slag drips on to the test specimen and flows over it and into a lower section of the furnace, from which it is removed during the test.

As shown in Fig. 48 the test walls are inclined at 15 deg. between 14-in.-square corner pillars, which must be made of good material in order to withstand the soaking heat on two sides. Attention is called to the action of molten slag on the test walls; also note that walls 27 and 30 are each built up of two kinds of brick, the lower brick being hand-made sillimanite, while the upper brick are of high-grade fireclay.

The standard slag used in the tests was of the porous variety obtained from the walls of the Springdale boiler furnaces, and

was crushed to pass a $\frac{1}{8}$-in.-mesh screen. A typical analysis shows it to contain 45 per cent silica, 35 per cent alumina, 14 per cent iron oxide, 5 per cent lime, and 1 per cent magnesia.

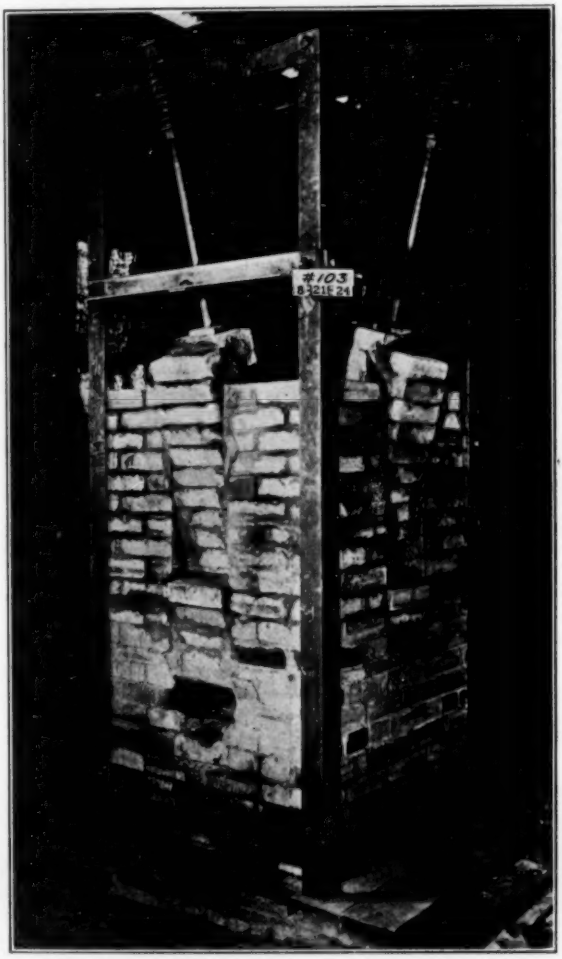

FIG. 46 SLAG FURNACE WITHOUT ROOF

In order to test the slag-eroding properties of coal from different sections of the country, two test runs were made with five samples of coal slag in which the iron content varied from 14 to 41 per cent. It was found that this iron content had a very marked influence on slag action on the furnace walls. While it is generally believed

that the fusion point of a coal ash is the determining factor in the rate of slag erosion, the fusion point is usually determined by the Seger cone, and is really the softening point, whereas it is necessary to liquefy the slag before it is detrimental to brickwork. It was found that some slags which have a low fusion point were really

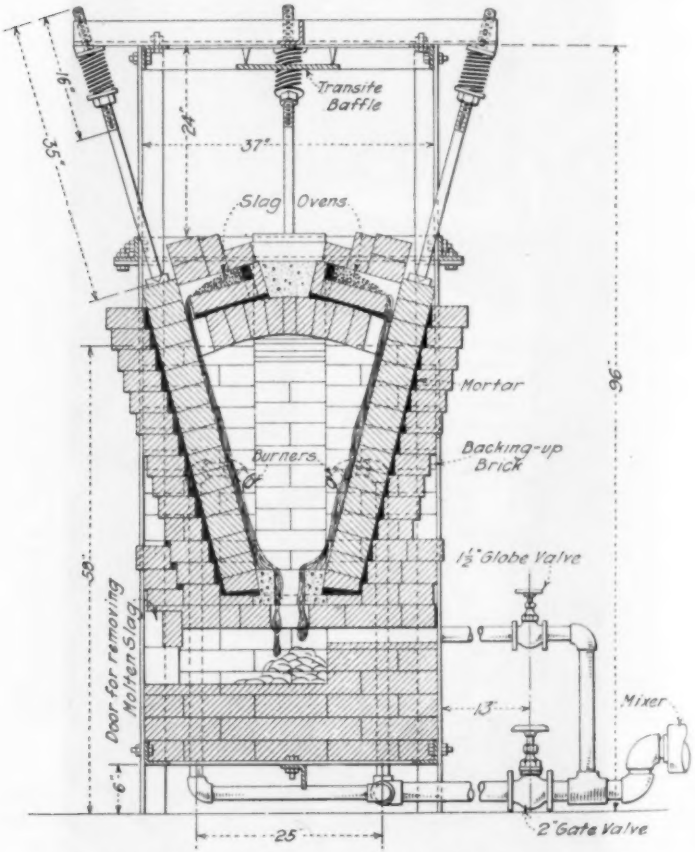

FIG. 47 SECTIONAL ELEVATION OF SLAG TEST FURNACE

the least detrimental, while others with apparently a high fusion point were very erosive.

It will be noted in Table 10, giving slag analysis, that specimen 5 which we found to be the most viscous and consequently least erosive, has a fusion point of cone 12, while specimen 3, which was the most erosive, has a fusion point of cone 13. Slag specimen 5

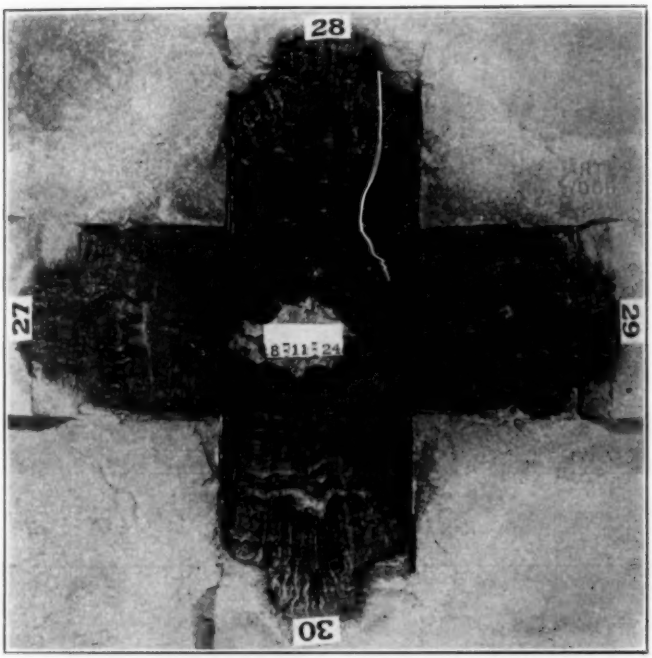

FIG. 48 VERTICAL VIEW INTO SLAG TEST FURNACE AFTER A RUN

had but 3 per cent of iron, while slag specimen 3 has over 40 per cent.

Fig. 49 shows part of the slag ovens in which the iron content of the slag used in *A* was over 40 per cent while that of *B* was about 14 per cent. Considering that *A* ran only 14 hours and that *B* ran

FIG. 49 SLAG OVENS AFTER RUN SHOWING EROSION

FIG. 50 CEMENT GUN IN OPERATION

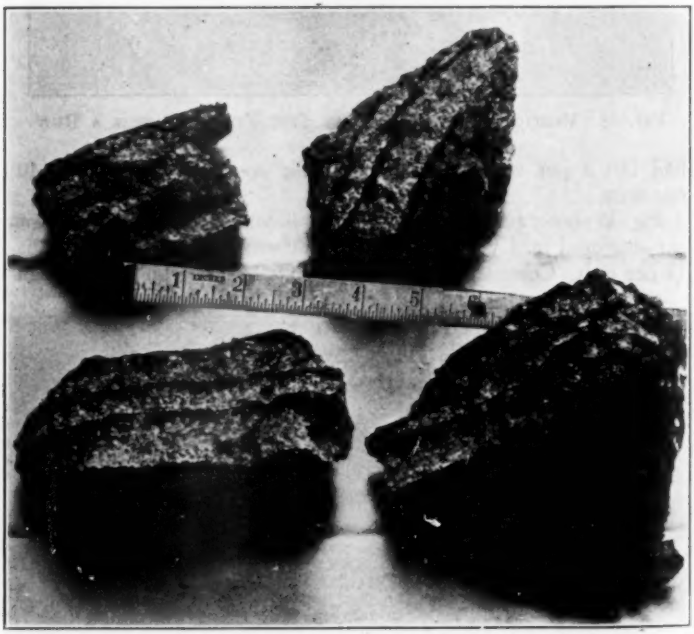

FIG. 51 SAMPLE OF PATCHING MATERIAL APPLIED WITH CEMENT GUN

TABLE 10 CHEMICAL ANALYSIS AND FUSION POINT OF COAL-ASH SLAGS USED IN FIREBRICK TESTS

	1	2	3	4	5
	Springdale	Keystone Power	Windsor	Potomac Edison	Detroit Edison
Silica	43.24	43.94	33.48	45.40	50.68
Alumina	33.57	34.97	19.08	38.19	39.77
Iron oxide	14.01	14.97	40.58	9.41	3.15
Titania	1.02	1.12	1.08	0.96	0.28
Lime	5.09	2.43	3.20	3.56	1.77
Magnesia	0.87	0.75	1.05	0.85	1.15
Alkalis	1.90	1.46	1.21	1.95	3.11
	99.70	99.64	99.68	100.32	99.91

Fusion Points				
1	2	3	4	5
Cone 13	Cone 15	Cone 13	Cone 17-18	Cone 12
1390 deg. cent.	1430 deg. cent.	1390 deg. cent.	1480 deg. cent.	1370 deg. cent.
2534 deg. fahr.	2606 deg. fahr.	2534 deg. fahr.	2696 deg. fahr.	2498 deg. fahr.

FIG. 52 TEST WALL AFTER RUN; HIGH-SODIUM-SILICATE MORTAR

18 hours, the effect of high iron content in coal ash is quite apparent.

The results of our investigation disclose the following:

a It was not possible to apply any coating to slag-eroded furnace walls except by means of a mechanical device in which the material is driven into pores of the wall. Fig. 50 shows material which is 94 per cent calcined and 6 per cent plastic, being applied with a cement gun. Fig. 51 shows a specimen of this material taken from a furnace wall, and it is clear that there have been four patches applied after successive runs, and without removing the slag coating.

b Mortar composed of a large percentage of calcined material

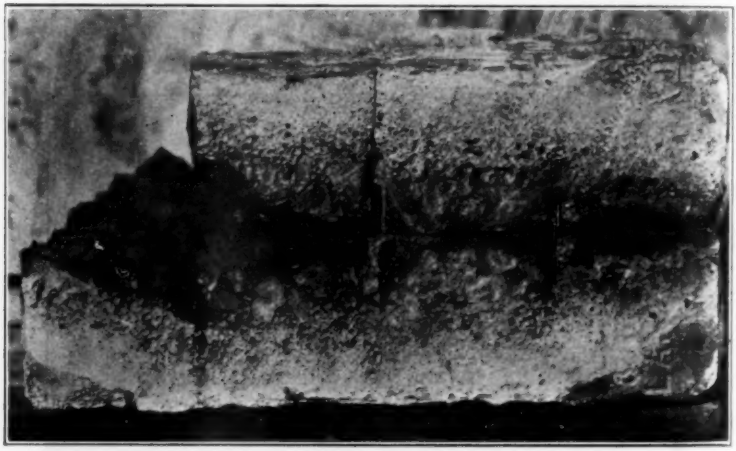

FIG. 53 SAME AS FIG. 52, BROKEN AT A JOINT

finely ground with high-fusion plastic clay makes a joint equal in strength and resistance to slag erosion to that of any of the high-priced trade cements, and this mortar is superior to most of them.

Fig. 52 shows a wall laid with a well-known trade cement having a large percentage of sodium silicate. Note how a temperature of 2850 deg. fahr. has caused the mortar to run out of the joints, which allowed the brick to sag. Fig. 53 shows this wall broken at the joint and laid back on itself. Fig. 54 shows a wall in the same run as Fig. 52 but in which the mortar is 75 per cent fireclay and 25 per cent calcined. By using 75 per cent calcined and 25 per cent plastic, the shrinkage cracks shown in this mortar were eliminated in later tests.

One cement, not so readily eroded by slag as the brick, was left as a dark fin projecting into the furnace when the brick was

eroded, and this appeared to have a detrimental effect on the furnace walls, probably due to the fact that this fin absorbed heat readily from the furnace gases, and gave up this to the slag flowing over it, which tended to increase the fluidity and solubility of the slag for the firebrick. However, it is possible that the increased erosion may have been partly mechanical and caused by the

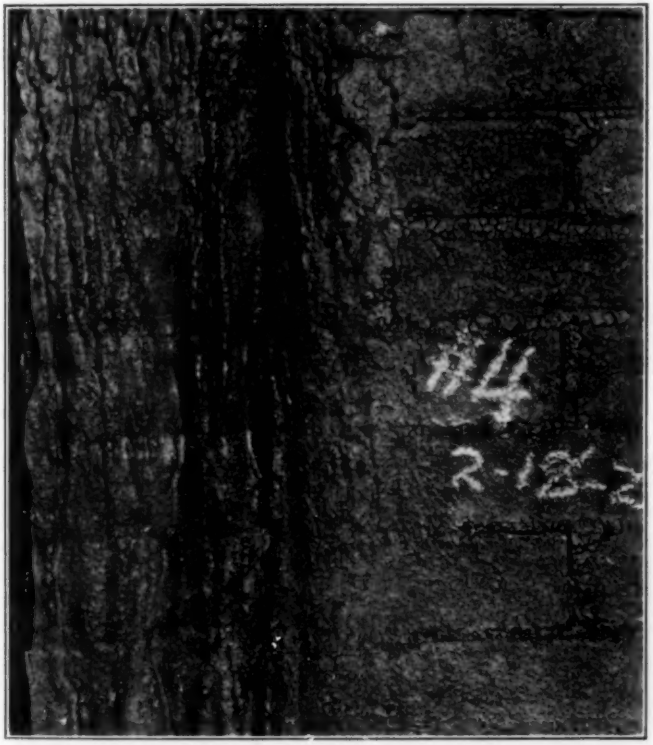

FIG. 54 TEST WALL AFTER RUN; MORTAR 75 PER CENT FIRECLAY AND 25 PER CENT GROG

impingement of the flame against the fins. This is illustrated in specimen 53 of Fig. 55 which is one of the four walls tested in the same run.

c It was found in a comparison of the various brick that:

1 Material being the same, the more dense the brick the greater the resistance to slag action. Fineness of grind, minimum moisture content and great pressure during manufacture all help to produce this resistance. Specimens 55 and 57 in Fig. 56 show walls with

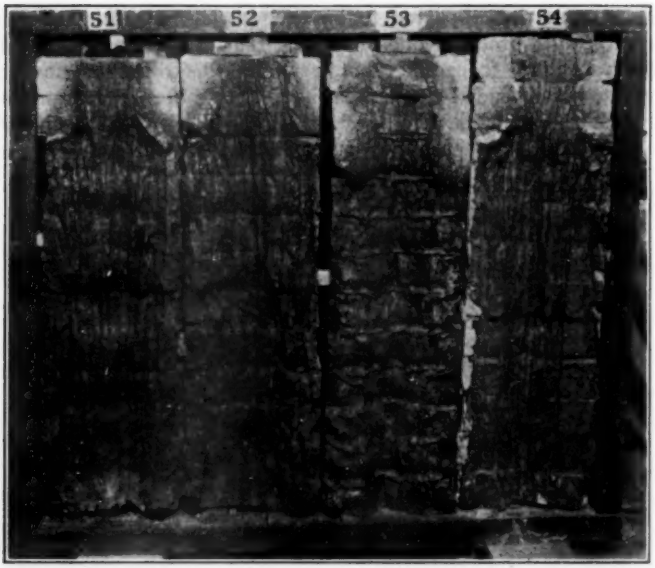

FIG. 55 TEST WALLS FROM SAME TEST AFTER RUN

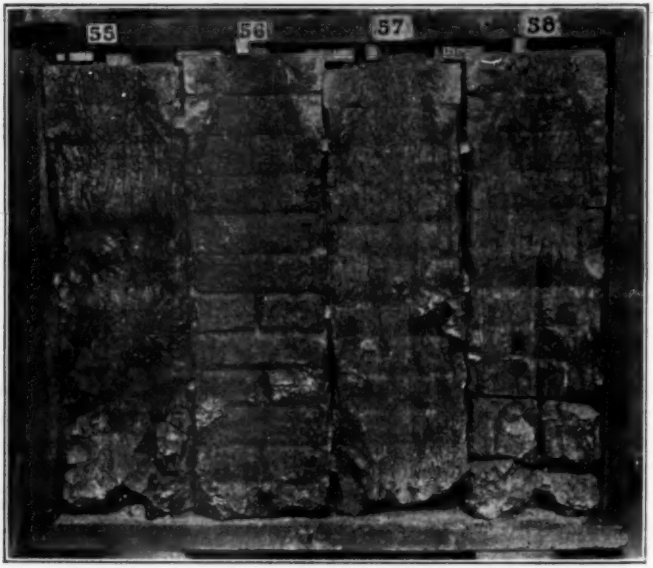

FIG. 56 TEST WALLS FROM SAME TEST AFTER RUN

brick made from the same high-grade flint and plastic clay, but 55 was a stiff mud repressed brick, while 57 was dry pressed.

2 Brick made of diaspore, calcined kaolin, or sillimanite are superior to clay brick in resistance of slag erosion. Specimen 56 is a diaspore brick, while specimen 54 shown in Fig. 55 is recalcined kaolin, and the sillimanite brick is shown in Fig. 48.

3 High-alumina brick suffer far less by slag erosion than other clay brick, but they are much more likely to shrink and crack.

4 Resistance to spalling and resistance to slag erosion are opposite properties in firebrick, and changes made in material or process of manufacture to prevent spalling seem to cause increased slag erosion, and vice versa. Test specimen 55 was a brick which in our corner pillars best withstood spalling and shrinking, but it can easily be seen that this brick suffered most from slag erosion, while specimen 56 failed completely in the corner pillars during its first run.

In conclusion, when the coal has ash with a high iron content, a brick made to resist slag erosion should be used, unless the furnace walls are kept below the temperature at which slag flows in liquid form; but if the temperature could be kept below this point by use of water backs, air cooling, or other means, then a brick of different composition having small tendency to spall would prove most satisfactory.

GEO. A. BALZ.[1] The paper under discussion is without doubt a most valuable contribution to the literature on a subject which, notwithstanding its fundamental importance, is rather sadly in need of the kind of information which the author has most ably presented in a very interesting and readable form. The paper is particularly valuable on account of the author's wide experience in the performance of refractory materials under various conditions of boiler-furnace service.

It is the desire of the writer to discuss the paper from the standpoint of the manufacturer of refractories and thus endeavor, if possible, to coördinate the two viewpoints; where such is not feasible, an effort will be made to emphasize the limitations to which the manufacturer is subjected by the consumer and by the raw materials available for the product.

The author stresses, and rightfully so, the fact that until comparatively recently the manufacture of refractories, in most cases, has been accomplished by methods that, in comparison with many of the industries which refractories serve, must be regarded as lacking scientific aspects to a decided degree. This condition, however, has not been brought about by any absence of interest

[1] President, Seaboard Refractories Company, Perth Amboy, N. J. Mem. A.S.M.E.

on the part of the manufacturer but largely by reason of the fact that the return secured by the manufacturer for his product has in most cases been insufficient to do more than reimburse him for actual outlays for raw materials, labor, and supplies. Overhead was, and in many cases still is, regarded as something that must somehow be met by any difference that may remain when deducting actual production cost from income from sales.

During the past fifteen years we have witnessed several large combinations and mergers of manufacturers of refractories, and it can be said that the realization of the operating and sales economies which these mergers have made possible is very closely coincident with the dawn of an era of scientific research and control in the industry.

The author refers to the methods adopted by some manufacturers for sizing the 9-in. brick for the purpose of assuring a more uniform product, so far as dimensions are concerned. Unfortunately, these methods thus far control mechanically only the thickness of the product; the length and width must still be subject to the variations produced by changing consistencies of the mud and the degree of burn of the finished product. The last-named variables are causing the manufacturer the greatest amount of difficulty, and are being made the subject of intensive study and research.

The author forcefully develops the influence of underburning upon the ultimate life of the firebrick. In most cases a well-burned product assures a reasonable period of service; there are several industrial applications, however, apart from boiler furnaces, where the installation of a hard-burned product would develop immediate failure of the refractory. Inasmuch as kiln firing is one of the major components of production cost, the consumer should not expect to secure a well-burned brick of uniform quality at a price that may be only slightly higher than that of common building brick. Neither should the manufacturer of a high-quality product be expected to compete with a brick that should rightfully be classed with face brick or perhaps pavers.

The analogy to which the author refers, between the development that has occurred in the manufacture of spark plugs and that required in the refractories industry, is interesting. It cannot be said, however, that the conditions existing in the respective industries are readily comparable. Heretofore the manufacturer of refractories, mostly through the methods and activities of the buyer, has been compelled to build his product to the price which the buyer has either mentally or orally set as the maximum, or which has been fixed by competition with the paving-brick variety of refractory. These conditions can hardly be said to have obtained in the spark-plug manufacturing industry.

Undoubtedly the development of refractories has to an appreciable extent been retarded by the lack of frankness and coöpera-

tion between the user and manufacturer. This has most generally been brought about by the consumer's apparent readiness to blame forthwith the refractories for any failure that may have occurred, or for any curtailment in the life of the material; thus disregarding the probable influence of changed furnace conditions, fuel, or any one of the factors to which the author draws attention. In the endeavor to prove his case, the consumer may have wilfully or tactfully failed to develop all of the facts involved, and thus an opportunity is lost to the manufacturer of the refractory to devise means for preventing a repetition of the difficulty. The average manufacturer of firebrick is an earnest, well-meaning, responsible, intelligent individual, desirous of placing the industry upon the high plane which, owing to the essentiality of its product to humanity and to all forms of industry, it undoubtedly merits.

The author states that plastic clays generally contain a higher percentage of impurities than the flints and semi-flints. Although this remark may be applied to plastic clays found elsewhere than in the New Jersey refractory-clay deposits, it cannot be said to apply generally to the latter. Some of these deposits are remarkable for their purity and high melting point with low drying and firing shrinkage. Their high degree of plasticity permits the manufacturer to produce a fairly dense brick, particularly when hard burned, without resorting to the admixture of clays of lower melting point for bonding purposes. These products are highly resistant to thermal shock and slag erosion. The ability of New Jersey brick to withstand heavy loads at the higher temperatures, without deformation, is perhaps too well known to require comment.

The fireclays of New Jersey should be classed entirely separate and distinct from the plastic clays found elsewhere in the United States. Their confusion with some of the lower-grade plastics frequently found in deposits in other localities has developed in the minds of some consumers a notion that New Jersey refractory products are inherently inferior to firebrick made from the more flinty variety of raw materials. The importance of the position that the Jersey products are gradually acquiring, in many types of furnaces, is a clear refutation of this erroneous conception.

In discussing the subjects of slag erosion and spalling of firebrick, the author has called attention to two of the most frequently encountered and major causes for the failure of refractories. Without doubt these two factors should receive the most careful study and research, particularly in connection with the chemical composition of slags and their chemical and metallurgical reaction with the refractory. The attention of designers of boiler furnaces should be emphatically drawn to the necessity of relieving the refractory, so far as possible, from all static and dynamic stresses. Under conditions of high temperature alone we are dealing with a material that is slowly changing its composition and structure, and as much freedom as possible should be permitted for the

development of such changes by the provision of adequate expansion clearances. This is particularly important in these days where refractory walls are frequently laid up dry or with joints of minimum thickness, made with a non-compressible substance.

The author draws attention to the apparent inability of refractories to keep pace with the ever-increasing demands made upon them in modern boiler operation. This must be conceded; however, in fairness to the manufacturer it should be stated that the demands have been aggravated with an incredible rapidity which did not permit an opportunity for endeavors to keep pace. Although failures of refractories sometimes occur within a few hours, it is unfortunately a fact that conclusive results from actual service trials sometimes require a period of months for their determination. It is the writer's opinion that a large proportion of the failures encountered with modern boiler-furnace refractories can be traced to pressure conditions in the atmosphere of the furnace. This permits the positive entrance of destructive gases and slags into the pores of the refractory at a time when the latter is in its most vulnerable condition, by reason of the soaking and penetrating heat to which it is subjected. In the days of over-grate drafts these conditions were obviously absent.

In concluding this discussion, the statement may be made that the manufacturers of refractories are now fully alive to the problems confronting them; they are anxious to coöperate with the consumer in mutual efforts to improve the quality of their product, and ask only a fair and reasonable return for their efforts.

They are dealing with a raw material that is concealed in the earth and subject to the usual variations present in materials of this character. The industry has for generations popularly been regarded as one in which the presence of intelligence has not been overabundant. If such has been the case, and it is by no means conceded, we may look forward with the utmost confidence to a marked change in the future.

C. F. Hirshfeld.[1] Most firebrick users spend their time telling the firebrick manufacturers what they think of them and of their product, and the firebrick manufacturer who desires to continue in business is hardly in a position to complain of practices which make the service on the refractories very much more severe. The writer does not mean to absolve the manufacturers of all guilt, nor to give the impression that the manufacturers always know all that there is to know about the subject, because they do not, but it is a fact that if those of us who use firebrick in such things as boiler furnaces would give a reasonable amount of consideration

[1] Chief of Research Dept., The Detroit Edison Co., Detroit, Mich. Mem. A.S.M.E.

to the peculiar properties of the material we are handling, we would not have so much trouble with these imperfect materials.

C. W. PARMELEE.[1] The size of the mortar joints in the masonry seems to have some significance. It seemed to the writer that the large joint, of course, would be a source of weakness, and he is wondering if sufficient attention has been paid to that item in construction. Of course, this takes us into the field of the mortars which should be used. That in itself is an extremely important point, because with the use of the higher-grade refractories more care and judgment will have to be used. Otherwise the use of the better grade will be offset by carelessness in masonry or carelessness in the selection of material.

In regard to the lack of progress that is being made in the manufacture of refractories, it is only within the last ten years that superior refractory material has come into use; formerly, it was sent to the dump. Today manufacturers are only too glad to work over and recover this important material. It offers a great field for research.

H. J. DONALD.[2] The question of joints was one which might well be mentioned. Of course, the writer is familiar only with furnaces using fuel oil, so the slag problem would not be important although we experience a little deterioration from sulphur. The average boilermaker will almost invariably spread mortar with a trowel and use it much thicker in consistency than we have found desirable at the Fuel Oil Testing Plant, Navy Yard, Philadelphia, where all the experimental work for firebrick for the Navy is done. We build about 150 tests walls a year for simulated tests, and we find that without much picking and choosing of the brick we get about $\frac{1}{32}$-in. joint. The mortar is made quite thin and a trowel is never used except for pointing up. The face of the walls in a simulated-service test are not washed at all, but the fireclay is washed off with a damp cloth so as to leave the brick face unprotected.

The mortar is made just thinner than can be handled with the trowel and the brick is dipped into or brushed through the mortar to remove air bubbles and to get a good coating. It is quite possible in setting up brick in ordinary furnaces, particularly with machine-made brick, made by stiff-mud or the dry-press processes, to have joints averaging an eighth of an inch. The average wall in a furnace is laid like a brick wall in a house. This is not correct, but it is quite a problem to teach the class of people who actually lay our walls to lay them with the proper joints.

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[2] Lieutenant Commander, U. S. Navy, Fuel Oil Testing Plant, U. S. Navy Yard, Philadelphia, Pa.

Another important factor is the care of the brick in handling. When the brick leave the manufacturers, they almost invariably have clean, sharp edges. By the time they have gone through the naval storehouse and two or three handlings on board ship, hardly a single brick will have satisfactory face for laying in a furnace. At least one company has adopted the practice of shipping brick in individual cartons, to protect them and treat them like face brick for buildings. This should do much toward eliminating the difficulties that begin with the joint.

F. A. Harvey.[1] Speaking from the manufacturer's standpoint, the manufacturer of firebrick has been afraid of specifications. A paper of this sort in its sane treatment of the subject will be a great aid in the formulation of specifications which the manufacturer can meet.

One point that was brought out, both in the paper and in the discussions following, was that resistance to slag action and resistance to spalling are contradictory qualities. We frequently find that specifications are written which involve both of these qualities. Frequently they are made so severe that it requires the best brick which can be produced to meet either one separately, and the combination is impossible.

The second point which the writer wishes to emphasize and which the author brought out, is that all the present laboratory tests for spalling are unsatisfactory. There are four that the writer knows of, and we frequently find brick in service which contradict the laboratory tests on spalling.

E. A. Brooks.[2] Is the writer to understand from the author's paper that an expansion joint would be generally advisable, inasmuch as most firebrick have a positive coefficient of expansion?

The writer would also like to know if it is not advisable to have an analysis of the bonding material (high-temperature cement) before accepting it, inasmuch as each coal has different percentages of iron, ash, and sulphur.

The Author. Mr. Thielscher brings out very clearly the seriousness which the refractory problem may assume in furnaces burning certain low-grade bituminous coals. He has given a definite numerical value to the loss in boiler-plant capacity chargeable to boiler-furnace troubles under conditions which he describes, and an intimation of the investment which this boiler-plant capacity may represent. Too often the expense of boiler-furnace maintenance is

[1] United States Refractories Corporation, Mt. Union, Pa.
[2] Brooks-Fisher Company, Atlanta, Ga. Mem. A.S.M.E.

gaged solely in terms of current labor and material costs. In stations operated on low annual load factor such costs are frequently a reasonably true measure, but under conditions of relatively high annual load factor and frequently in hydraulic relay service even though the annual load factor be low, boiler-furnace refractory troubles are a direct charge upon plant capacity and the plant investment must be increased to provide the reserve capacity required.

Mr. Hankison has developed a remarkably effective type of experimental furnace; he is to be congratulated upon having arrived at a design that permits on laboratory scale such close simulation of actual service conditions. His comparison of erosive effects and cone-fusion temperatures of different slags emphasizes the difficulty of interpreting the results of the ordinary cone-fusion test of slag without the aid of other data. Mr. Hankison's summary of experience in patching slag-eroded areas of boiler-furnace walls outlines a simple and comparatively inexpensive method of repair which has proved effective in many instances; this form of repair may often be applied to advantage where conditions are not too severe.

Mr. Balz and Dr. Harvey have contributed most valuably to the discussion in presenting the firebrick manufacturer's point of view. Unquestionably the failure of the firebrick industry to keep pace with boiler-furnace-refractory requirements cannot be laid to the manufacturer alone; as Mr. Hirshfeld points out, the user has rarely shown a practical interest in assisting toward the solution of the problem. Mr. Balz calls attention to the distinctive qualities of the New Jersey clays. This is a matter which may very appropriately be stressed, as the exceptional purity of a number of the higher-grade plastic clays of New Jersey is not generally recognized.

Professor Parmelee and Commander Donald emphasize the importance of the mortar joint as a factor in the life of the firebrick wall, and Mr. Brooks inquires as to the use of high-temperature cements and the advisability of special provision for thermal expansion. The thick mortar joint of raw clay has been a frequent source of refractory failure, both on account of the parting at the joint resulting from shrinkage of the clay, and because of the commonly lower refractoriness of the plastic clays which are often a large constituent of such mortars. For severe service conditions the joints should be thin and the mortar employed should be composed very largely of calcined material. The importance of mortar of this character is also mentioned by Mr. Hankison.

Answering Mr. Brooks' questions: Expansion joints of some type are generally advisable in the modern boiler-furnace wall. Some years ago when furnaces were of small dimensions and raw-

clay mortar was used almost exclusively, the important of special provision for expansion was not so great; the shrinkage of the mortar took up a large part of the expansion; but, even under those conditions, the walls were subject to distortion and frequently cracked badly. Many high-temperature cements contain sodium silicate, added to give cold-setting qualities, and other ingredients which under some conditions may act as a flux. Scientific care in selection of the mortar is practically as important as in selection of the firebrick.

No. 1984

AUXILIARIES FOR MOTOR VESSELS

By John W. Morton,[1] Camden, N. J.
Member of the Society

and

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Non-Member

With the vast increase in motorships it is only natural to expect a variety of installations; and the many designs and variety of types of propelling machinery and modes of drive in turn affect the auxiliary equipment.

The great differences that exist in auxiliaries in such factors as weight, space, duty, capacity, cost, etc., without doubt could be minimized.

Determining the right kind and capacity of auxiliaries, in order to reduce first cost and weight and at the same time have the most economical marine plant both at sea and in port, is a somewhat difficult task, because one has to consider the ship's trade routes, and whether the ship is new or converted.

It must also be remembered that the assumption regarding continual running with full cargo is rather ideal, but as all ships will be equally liable to runs in ballast or with part cargo or load, the results will be relative.

Comparisons between types of machinery are often based on the relative space required, the weight and fuel-consumption values, and the resulting saving in fuel.

While perhaps these are quite fair to the machinery, they are not to the ship as a whole, and are apt to show differences, the benefits of which in practice are never realized.

The comparisons in the paper have been made without bias for or against any particular type of machinery, and in a manner that shows a more true ultimate saving than that attained by other comparative methods.

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[2] Guarantee Engineer, N. Y. Shipbuilding Corp.

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The differences brought out by the analysis, although showing no one particular type as vastly superior, do show that certain vessels can be built and operated with more profit than others, and they therefore justify a close investigation of ships' economies when selecting the type of machinery.

PART I

By J. W. Morton

Steam Versus Electric Auxiliaries

DUE TO variations in types of ships and propelling engines, the necessary auxiliaries vary to such an extent that standardization is almost impossible. This of course does not mean that within the machinery itself or in its component parts standardization is impossible, as manufacturers are striving today more than ever to standardize their products as much as possible, greatly to the benefit of shipowners or operators. But auxiliaries on board a ship will vary according to the character of the ship's service, whether tramp, passenger liner, tanker, barge, towboat, dredge, pleasure vessel, etc. Each ship is likely to have a different kind of installation. The mode of drive, whether direct, electric, hydraulic, or gear-transmission, and whether or not the ship is new or converted, are factors influencing the installation.

2 To the shipowner who has decided to install Diesel engines as the propelling unit in his vessel, the selection of suitable auxiliary machinery is a question worthy of special consideration and presents greater difficulties with oil-engined vessels than with steamers. With the latter, either the donkey boiler or the main boiler supplies all the steam necessary from the capstan aft to the windlass forward, and all other auxiliary machinery in between.

3 When the appalling inefficiencies of the steam-driven windlass, winch, steering gear, capstan, etc., are considered, one wonders why such extravagance is still permitted. The simplicity of steam auxiliaries and their reliability probably account for their use. Surely a well-tested electric auxiliary capable of working in the open and of proven reliability should make steam auxiliaries obsolete.

4 In vessels above, say, 2000 tons deadweight capacity, it is advisable to arrange the steam plant in two or more units, in order to allow repairs or examinations to be carried out without shutting down the whole plant. In special cases, such as oil carriers, where steam is required almost continuously for tank heating, etc., steam auxiliary machinery will not readily be displaced, but a combination of steam and electrically driven auxiliaries is a very satisfactory alternative.

5 Much has been written of the advantages of oil-engined vessels over those fitted with steam engines, but little has been said about the large amount of auxiliary machinery to be found in a large oil-engined ship.

6 Today, the electrically driven auxiliary machinery now available for marine work is in every respect as reliable in service as the best type of steam plant, and possibly calls for less attention to maintain it at maximum efficiency. For cost of operation of either type of installation, reference is made to Tables 1 and 2 giving hypothetical cases, estimated by Mr. Robert Love.[1] He further states:

> The power requirements (Tables 1 and 2) of the auxiliary machinery have been estimated on a basis of the horsepower of each machine, obtained from the results of trials of electrically driven units. In the case of the steam-operated auxiliary machinery the required horsepower of each machine has been taken as being the same as for similar electrically driven gear and the steam required at an average rate of 60 lb. per horsepower per hour.

7 Steam machinery for engine-room and deck purposes can be relied upon to give continuous service under sea-going conditions with low upkeep cost, but in vessels propelled by Diesel engines it should be observed that even when the most economical arrangement and type of steam machinery is fitted, the fuel required for auxiliary purposes at sea will amount to from 30 to 35 per cent of that used by the main propelling machinery, or approximately 25 per cent of the total fuel per day at sea.

8 With electric auxiliaries using current generated by Diesel-driven generators, the fuel required for auxiliary purposes at sea will not exceed 5 per cent of that required by the main engines, or approximately 4.5 per cent of the total fuel per day at sea Also, when working cargo, the electric winches perform the same work as steam winches with a saving of approximately 90 per cent of the fuel.

9 The electrical auxiliary machinery has a higher initial cost. It should be observed, however, that against this must be set the cost of the extra fuel continuously required if steam plant is used. In a very few years' service the additional first cost of the electric plant will be completely written off. Another point not to be forgotten is the lubricating-oil consumption. High-speed steam engines are as a fact very economical in the use of lubricants. Of course each individual case should first be thoroughly analyzed from all angles.

10 In the past, an electric plant in sea service has been considered to require much more attention than steam gear, calling for special knowledge outside the scope of the average sea-going engineer. That this is not now the case is clearly proved by the

[1] Transactions, Institute of Engineers and Shipbuilders in Scotland, vol. 47, part 2, December, 1923, p. 121.

large number of such installations in completely successful service. For use on board ship, none but the best class of electric gear should be fitted. Motors and control gear for deck use should be of the

TABLE 1 BRAKE-HORSEPOWER REQUIREMENTS OF A SINGLE-SCREW, DIESEL ENGINED VESSEL, 2000 TONS D.W.C., 1100 B.HP., 10.5 KNOTS

WITH STEAM AUXILIARY MACHINERY

Auxiliary	Equivalent b.hp. maximum output	Normal sea load	Maximum maneuvering load	Port load—working cargo	
				Maximum	Mean
Steering gear	8	2	3	..	..
Six winches	16(ea.)	..	..	64	48
Windlass	50	..	30	..	..
Heaters	20	7	15	15	6
Lighting	7.5	5	5	7.5	5
Auxiliary compressors	80	..	65	..	..
Jacket pumps	7	6	6	..	..
Piston-cooling pumps	4	3	3	..	..
Lubricating-oil pumps	4	3	3	..	..
Bilge and sanitary pumps	10	6	6	6	6
Ballast pump	22	..	..	..	..
Fuel-oil pump	1.5	..	..	..	..
Maneuvering gear	4	..	4	..	..
Turning gear	7	..	..	7	..
Total b.hp.		32	140	99.5	65
Taking steam consumption of auxiliary mach. at 60 lb. per b.hp-hr., lb.		1920	8400	....	3900
Oil used per hr. at 12 lb. of steam per lb. of oil		160	700	....	325
Oil used, tons per day		1.71	....	....	3.5

Fuel for main engine, 4.75 tons per day
Total fuel, main and auxiliary machinery at sea, 6.46 tons per day
Fuel used in port when working cargo, 3.5 tons per day

WITH ELECTRICALLY DRIVEN ENGINE ROOM AND DECK AUXILIARIES CAPABLE OF THE SAME OUTPUT

	Normal sea load	Maximum maneuvering load	Port load—working cargo	
			Maximum	Mean
Total b.hp.	32	140	99.5	65
Actual loads	110 amp. at 220 volts	330 amp. at 220 volts	..	..
Oil used per hr. by Diesel gen., lb.	17	70	..	32.5
Oil used, tons per day	0.183	..	..	0.35

Fuel, main engine, 4.75 tons per day
Total fuel, main and auxiliary machinery at sea, 4.933 tons per day
Fuel used in port when working cargo, 0.35 tons per day

Days at sea, 250	Vessel with steam-driven auxiliary machinery Tons fuel	Vessel with electrically driven auxiliary machinery Tons fuel
Main engine	250×4.75 = 1190	250×4.75 = 1190
Auxiliary machinery	250×1.71 = 428	250×0.183 = 46
Days working cargo, 40	40×3.5 = 140	40×0.35 = 14
Days in port, 75	75×0.75 = 56	75×0.095 = 7
Total fuel for one year, in tons	1190 for main engines 624 for aux. engines	1190 for main engines 67 for aux. engines
Cost of fuel per year, based on prices ruling at Aug. 1925	1190 tons at $14.70 per ton = $17,493.00 624 tons at $11.20 per ton = $6,988.80	1257 tons at $14.70 per ton = $18,477.90
Saving in fuel cost per year by using electric auxiliaries	$6003.90	

watertight enclosed type, special attention being given to the rating of the motors and resistances in view of the high temperatures in which operation may be required, as, for instance in the tropics.

11 In the case of cargo winches, which are generally handled by unskilled labor such as is available in the ports visited, it is amazing to see the treatment which winch control gear may have to endure. Further, the method of handling cargo varies in different ports.

TABLE 2 BRAKE-HORSEPOWER REQUIREMENTS OF A TWIN-SCREW, DIESEL-ENGINED VESSEL, 10,000 TONS D.W.C., 2250 B.HP., 10.5 KNOTS

WITH STEAM AUXILIARY MACHINERY

Auxiliary	Equivalent b.hp. maximum output	Normal sea load	Maximum maneuvering load	Port load—working cargo: Maximum	Port load—working cargo: Mean
Steering gear	20	4	5	...	..
Nine winches	27(ea.)	..	..	135	80
Windlass	..	..	50	...	..
Heaters	30	10	20	20	8
Lighting	10	7	7	10	7
Aux. comp., 2 fitted each 53 hp.	106	..	80	...	..
Jacket pumps	15	12	12	...	..
Piston-cooling pumps	7	6	6	...	..
Lubricating-oil pumps	7	6	6	...	..
Bilge and sanitary pumps.	12	8	8	8	8
Ballast pump	25	..	..	...	..
Fuel-oil pump	3	..	..	...	..
Maneuvering gear	10	..	8	...	..
Turning gear	14	..	..	7	..
Total b.hp.		53	202	180	103
Taking steam consumption of auxiliary mach. at 60 lb. per b.hp-hr., lb..	3180		12120	....	6180
Oil used per hr. at 12 lb. of steam per lb. of oil		265	1010	....	515
Oil used, tons per day		2.84	....	....	5.53

Fuel for main engines, 9.75 tons per day
Total fuel, main and auxiliary machinery at sea, 12.59 tons per day
Fuel used in port when working cargo, 5.53 tons per day

WITH ELECTRICALLY DRIVEN ENGINE ROOM AND DECK AUXILIARIES CAPABLE OF THE SAME OUTPUT

	Normal sea load	Maximum maneuvering load	Port load—working cargo: Maximum	Port load—working cargo: Mean
Total b.hp.	53	202	180	103
Oil used per hr. by Diesel gen., lb.	28	100	...	51
Oil used, tons per day	0.3	...	...	0.545

Fuel, main engines, 9.75 tons per day
Total fuel, main and auxiliary machinery at sea, 10.05 tons per day
Fuel used in port when working cargo, 0.545 tons per day

	Vessel with steam-driven auxiliary machinery Tons fuel	Vessel with electrically driven auxiliary machinery Tons fuel
Days at sea, 200		
Fuel for main engines	200×9.75 = 1950	200×9.75 = 1950
Fuel for aux. engines	200×2.84 = 567	200×0.3 = 60
Days working cargo, 50	50×5.53 = 276.5	50×0.545 = 27.7
Days in port, 115	115×1.5 = 173	115×0.15 = 17.3
Total fuel for one year, in tons	1950 for main engines 1016.5 for aux. engines	1950 for main engines 105 for aux. engines
Cost of fuel per year, based on prices ruling at Aug. 1925	1950 tons at $14.70 per ton = $28,665 1016.5 tons at $11.20 per ton = $11,384.80	2055 tons at $14.70 per ton = $30,208.50
Saving in fuel cost per year by using electric auxiliaries..	$9841.30	

These points must receive the careful consideration of the designers of deck machinery.

12 Direct current is in general used on board ships now, both 110 and 220 volts, the 220-volt for power purposes and the 110

for lighting, obtained by a motor-generator set and balancer system. With high voltage a considerable saving in cost and weight of cables and motors is obtained. It is granted that special attention must be given to insulation where higher voltages are used, but the advantages due to the reduction in the size of cables, etc., are substantial, and merit the careful consideration of the various classification societies.

All-Electric Auxiliaries

13 In vessels with all-electric auxiliary machinery, it has been found, except in very special cases, that three Diesel-driven generators is the minimum number fitted, the output of each generator being so proportioned that two engines will be capable of handling the maximum current demand under maneuvering conditions. This insures that under normal sea conditions, one generator, operating at from one-half to three-quarters load, will suffice, and when working cargo one or two generators may be used as necessary. With this arrangement, one generating set can always be under periodic examination or overhaul, thus insuring that the whole plant is kept in the most efficient condition.

14 The auxiliary generating sets of the direct-connected type generally run at from 300 to 400 r.p.m. Such speeds give comparatively light engines and dynamos, but in service, the engines require a large amount of attention. Choosing the right kind of auxiliary sets for the particular type of installation, whether two- or four-stroke, single- or multiple-cylinder, high-, medium-, or low-speed type of engine, is a personal problem with the shipowner or operator, but the author's opinion is that a moderate-speed, multiple-cylinder type of direct generating set as an auxiliary will give the least trouble, due to better operation over a long period with absolute reliability, lower number of reversals, less vibration, lower lubricating-oil consumption, and low up-keep cost. The engine of moderate speed occupies more space and is heavier than one of the high-speed type for equal powers, but the more satisfactory performance and reduced up-keep cost in service outweigh these disadvantages.

15 Dynamos are generally of the open, splash-proof type; all other motors in the engine room are likewise of this type, or, as is the case with the motors for deck machinery, of the enclosed, ventilated type.

16 The electric-power requirements of all sizes and types of merchant vessels, other than the largest passenger ships, can generally be satisfied by a combination of standard-sized cylinder units. One should consider the alternative methods of powering or driving the auxiliaries, the methods available, the advantages and disadvantages, and notes concerning their installation and operation. In case of steamships using steam power for main

engine and auxiliaries, this represents no difficulty. If powering by oil engine is adopted it means:

1 A separate engine for each machine, or
2 A generating station in the main or auxiliary engine room for the whole ship, or
3 A combination of (1) and (2).

The separate engine is practicable for each air compressor, winch, capstan, windlass, pump and the electric light, but not as convenient for the steering gear.

Winches

17 Fig. 1 shows method of driving the winches from an oil engine located in the deck house between hatchways, suitable for driving, say, four to six winches. How economical such a drive is, it is hard to say, but so much shafting, bearings, and gearing surely entails a loss and inefficiency exists in clutches and reversing gear. Such a central drive would have to be operating continuously, and one cannot say that all four or six winches will be hoisting, lowering, and stopping at the same moment. Further, the oil engine will have to be large enough to drive all the winches at any moment, although only one winch might be working. Other drawbacks to such an installation are the isolation of the units, the general overall inefficiency compared to a "central station," the high initial cost and increase in cost of installation due to necessary fuel, oil, and water-cooling connections, and, in winter time, the necessity of draining the jackets to prevent freezing. The advantages of a central station and powering by electricity are the great convenience and flexibility and the fact that practically all the machinery is under the eye of the engineer in the engine room.

18 Oil power has not the same flexibility as steam power, but improvements are always forthcoming and there is an attempt to do away with steam as far as possible in an internal-combustion-engined vessel. With an oil-engined ship, boilers are undesirable, but if motorships are equipped with boilers for auxiliary purposes it is worth while to make use of steam instead of adding other machinery.

Use of Compressed Air

19 Air has been used to a certain extent as an auxiliary-power medium, especially for the steering gear, while experiments have been made on deck winches too. An incorrect analogy between steam and air is often drawn. In the formation of steam it is a well-known fact that the principal amount of heat is required to overcome the latent heat. It takes only 2 to 3 per cent more heat to generate steam at 100 lb. pressure than at 10 to 20 lb. pressure, and the higher pressure is justified as it costs practically no more. With air, on the other hand, every extra pound of

pressure demanded over the pressure actually required means using so much extra power in driving the compressor. The advantage lies in using air at a low pressure. From an experiment carried out in England and on a ship fitted with 5-ton steam winches, it was found that the weight of the lift varied from 700 to 1400 lb. By putting a pressure gage in place of the drain cock on the valve chest of the winch, and by wire-drawing the stop valve on the winch, it was possible to ascertain the pressure necessary to operate the winch, and it was found that 15 to 20 lb. pressure was sufficient, even at full speed. The percentage of heavy lifts in dealing with cargoes is very small.

20 A simple low-pressure air compressor in the engine room could be used, and the air taken to the winches at 25 to 30 lb. It need not be more. The losses due to leakage are very much less than with steam. Only one pipe is needed from air compressor to winch because no return or exhaust pipe is needed. If suitable arrangements are made by gearing, then on the few occasions that a heavy lift is required, 20 lb. can still be used, and the load lifted more slowly.

21 Obviously the ordinary steam steering gear will operate just as well with air as with steam. Consequently the same gear with which every one is acquainted can be used. The quantity of air used for steering is surprisingly low. Messrs. Roylton, Dixon & Company (England) in 1913 made some experiments in regard to this in a 5000-ton ship, the *Berwindvale*, during her trials and on her voyage to New York and back. On trials on the measured mile there was, of course, a considerable amount of air used, but for an extremely short space of time, while the vessel was being put hard over. But on the run home the quantity of air used to keep the vessel on her course was extremely small, only 20 cu. ft. of free air per min., and the power required was very small.

22 Going to America and back, in fine weather with a skilful helmsman — and there was quite a large difference between a careless and a skilful one — under good conditions and in fair weather the quantity of air used did not exceed more than 35 to 40 cu. ft. per min. In heavy weather during eight hours of steady observations, the average quantity required was more than 90 to 95 cu. ft. per min. The results of this investigation led to the installing of this type of steering gear in an all-motor ship, the *Rolandseck* (sunk during the war), built by Tecklenborg Co. and engined by Carels. She had a single-stage air compressor driven by levers from the main engine.

23 The *Wotan*, a single-screw ship fitted with a 2000-hp. Carels engine, was equipped with the same form of steering-engine compressor, and it was highly satisfactory.

24 In some motorships, such as the *Ashbee* and the *Jacksonville*, several steam pumps for intermittent service will be connected to an air line, the air being generally supplied by bleeding

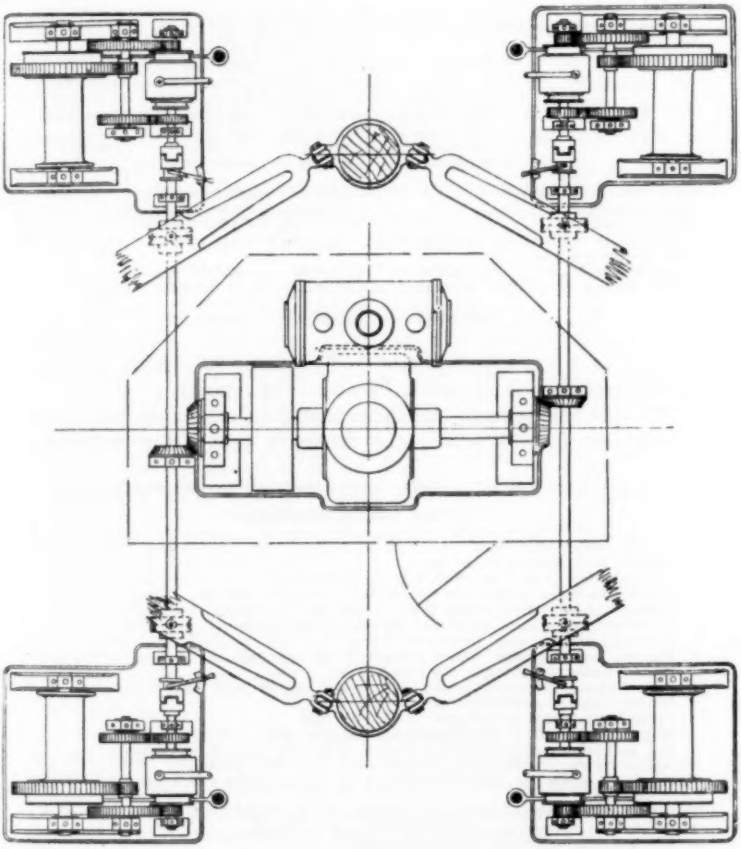

FIG. 1 METHOD OF DRIVING WINCHES FROM AN OIL ENGINE LOCATED IN THE DECK HOUSE BETWEEN HATCHWAYS

(Reproduced from Internal Combustion Engines, published by the Institute of Marine Engineers, London, 1922.)

the starting-air tanks through reducing valves. Another novel way of driving the deck machinery is found in a vessel of about 2000 tons dead-weight capacity and propelled by a Werkspoor Diesel engine. The owners, thinking that the usual alternative schemes available for driving auxiliary machinery were either very costly,

in the case of Diesel-engine-driven dynamos with electrically driven engine-room auxiliaries, winches, etc., or wasteful, as in the steamship with steam-driven donkeys and deck machinery, finally arranged a compromise layout which has effected excellent results,

TABLE 3 COMPARISON OF COSTS WITH VARIOUS TYPES OF AUXILIARIES

	Vessel with steam winches and steam auxiliary machinery	Vessel with steam aux. mach., uniflow eng., together with deck shafting for driving winches	Vessel same as described in previous column but with Diesel-eng.-driven deck shafting in lieu of uniflow eng.	Vessel with electrically driven aux. and electric winches
Days at sea, 250				
Main eng. and dynamo..	.....	1200 tons fuel	1200 tons fuel	.....
Days working cargo, 40.	.....	$40 \times 1.7 = 68$ tons	$40 \times 0.5 = 20$ tons	.....
Days in port, 75.....	.....	$75 \times .75 = 56$ tons	$75 \times 0.095 = 7$ tons	.....
Cost of fuel per year				
Sea service	$17,493.00	$17,640.00	$17,640.00	.....
Port service	6,988.80	1,388.80	396.90	.....
Total	$24,481.80	$19,028.80	$18,036.90	$18,477.90

both as to low production cost and economy in service, the ship having now been in operation for about three years.

25 The engine-room auxiliaries and deck machinery are steam driven, but for sea service the necessary pumps for circulating

TABLE 4 LIST OF AUXILIARY MACHINERY IN MOTORSHIP *CUBORE* BEFORE RECONDITIONING

No.	
2	100-kw. Diesel generators
1	100-kw. turbo-generator (steam)
1	Scotch boiler, 12 ft. 6 in. diam. x 11 ft. 0 in. long
2	Circulating-water pumps, twin centrifugal, 500 gal. per min.
1	Lubricating-oil pump, rotary plunger, 100 gal. per min.
1	Lubricating-oil pump, horizontal duplex, 8½ in. x 6 in.
1	Fuel-oil transfer pump, horizontal duplex, 5 in. x 6 in.
2	Fuel-oil supply pumps, rotary plunger, 8 gal. per min.
1	Ballast pump, centrifugal, 500 tons per hr.
1	Ballast pump, horizontal duplex, 10½ in. x 12 in., 200 tons per hr.
1	Sanitary pump, centrifugal, 50 gal. per min.
1	Fresh-water pump, centrifugal, 75 gal. per min.
1	Bilge pump, vertical centrifugal, 500 gal. per min.
1	Fire and bilge pump, horizontal duplex, 12 in. x 8½ in. x 12 in. (steam)
1	Sanitary pump, horizontal duplex, 6 in. x 4 in. x 6 in. (steam)
1	Combined condenser, air, and circulating pump, 12 in. x 14 in. x 14 in. x 12 in. (steam)
2	Starting-air compressors, duplex, 2-stage, 350 lb., 11¾ in. x 5 in. x 10 in.
1	Booster air compressor, duplex, 1200 lb., 1¾ in. x 6 in.
2	Refrigerating machines, 1 ton, ammonia
1	Donkey boiler, vertical, 3 ft. 0 in. dia.
1	Boiler-feed pump, vertical duplex, 7½ in. x 5 in. x 10 in. (steam)
2	Boiler fuel-oil pumps, horizontal duplex, 4½ in. x 2¾ in. x 4 in. (steam)
1	Windlass, 9 in. x 9 in. (steam)
6	Cargo winches, compound geared, 8½ in. x 10 in. (steam)
6	Mooring winches, 10 in. x 12 in. (steam)
1	Capstan, 8½ in. x 8 in. (steam)
1	Steering gear, screw type, 35-hp motor.

cooling water through the main-engine cylinder jackets and pistons for bilge purposes, boiler feed, condenser, etc., are all driven directly from the main engine. Steam is generated by passing the engine exhaust gases through two Scotch boilers, each having 430 sq. ft. of heating surface, the maximum working pressure

being 150 lb. per sq. in. A pressure of 110 to 120 lb. per sq. in. can thus be maintained, and the steam drives the steering gear and heats the saloon and crew's quarters. When entering or leaving port the boiler burners are lighted, and sufficient steam is immediately available for driving the auxiliary air compressor, windlass, etc. Electric light is supplied by a small dynamo driven by a semi-Diesel engine of 20 hp. which consumes about 1/20 of a ton of oil per day. The steam-driven auxiliaries are as follows: one simplex ballast pump, with a capacity of 130 tons per hour, one simplex feed pump for the boilers, and one simplex fuel-transfer pump. The boiler burners are fed by an efficient pneumatic device. For handling cargo in port a special arrangement has been adopted. A two-cylinder uniflow engine of 160 i.hp. is provided, which drives two lines of shafting through disengaging clutches, one going

TABLE 5 LIST OF AUXILIARY MACHINERY IN MOTORSHIP *CUBORE* AFTER RECONDITIONING

No.	
3	100-kw. Diesel generators
2	Circulating-water pumps, centrifugal, 500 gal. per min.
2	Combined lubricating- and fuel-oil service pumps, rotary plunger, 150 gal. per min. and 8 gal. per min.
1	Fuel-oil transfer pump, horizontal duplex, 5 in. x 6 in.
1	Aux. fuel-oil transfer pump, rotary plunger, 20 gal. per min.
1	Ballast pump, centrifugal, 500 tons per hr.
1	Ballast pump, horizontal duplex, 10½ in. x 12 in., 200 tons per hr.
1	Sanitary pump, centrifugal, 50 gal. per min.
1	Fresh-water pump, centrifugal, 75 gal. per min.
1	Bilge pump, vertical centrifugal, 500 gal. per min.
1	Fire pump, 2-stage centrifugal, 500 gal. per min.
2	Starting-air compressors, duplex, 2-stage, 350 lb., 11¾ in. x 5 in. x 10 in.
1	Booster air compressors, duplex, 1200 lb., 3¼ in. x 6 in.
1	Emergency air-compressor unit
2	Refrigerating machines, 1 ton, ammonia
1	Lubricating oil separator
1	Donkey boiler, vertical, 3 ft. 0 in. dia.
1	Heating-water circulator, centrifugal, 50 gal. per min.
1	Windlass, 75-hp. motor
6	Cargo winches, 25-hp. motors
8	Mooring winches, 50-hp. motors
1	Steering gear, screw type, 35-hp. motor

forward and the other aft. At the forward part of the ship there are six 5-ton winches and a windlass, and at the after part three 5-ton winches. The shafting is provided with flexible couplings to allow for the curvature of the deck, and with special bevel gears encased in cast-iron grease boxes for driving the athwartship lines to which the winches are coupled.

26 The shafting takes up no more space than the usual winch steam and exhaust pipes. After 18 months' use the teeth of the bevel gears were quite bright and showed no measurable wear. The windlass and each winch is provided with a friction clutch of special construction, designed to comply with the various conditions met on shipboard, and the measured average fuel consumption is 156 lb. per hr. or about 1.7 tons per day. The initial cost of this system is higher than for steam winches, but it is much more economical in steam consumption, an important consideration where oil fuel is used for raising steam. If a Diesel

engine were substituted for the uniflow steam engine, a further substantial reduction of fuel consumption would result. Assuming the same fuel consumption for the main engine as per Table 1 and quoting other figures from the examples given,[1] the difference in fuel expenditure for the various systems of auxiliary machinery is shown in Table 3.

27 As oil tankers are in a class by themselves, and a large amount of steam is needed for cargo heating at sea, it is only natural to use steam auxiliaries. Conversion of existing steamships to motorships presents also a special problem; and whether to retain all steam auxiliaries or scrap them and install electric auxiliaries, or to retain the existing deck machinery, except the steering gear, and electrify all engine-room auxiliaries, is a question which is sometimes rather difficult to decide. It is largely determined by the type of vessel and the kind of trade in which it is to engage.

TABLE 6 OIL CONSUMPTION AND SAVING DUE TO CHANGE TO ELECTRIC AUXILIARIES, MOTORSHIP *CUBORE*

	Oil consumption			
	Steam auxiliaries		Electric auxiliaries	
	Tons	Barrels	Tons	Barrels
At sea				
Boiler	58.143	407	.....	...
Aux. Diesel gen.	18.429	129	21	147
In port				
All auxiliaries	17.285	121	2.285	16
Total	93.857	657	23.285	163
Saving	.....	...	70.572	494

Fuel Consumption

28 A few examples of diverse ships, operating on different routes will show the variety in fuel-oil consumption.

29 The motorship *Cubore* (ore carrier) originally had several electric motor-driven engine-room auxiliaries and an electric steering gear, all other engine-room and deck machinery being steam driven with steam supplied from a donkey boiler. After a few years' operation the main engines were altered and at the same time the entire steam plant was removed and electric-motor-driven auxiliaries installed throughout. Tables 4 and 5 show the auxiliaries before and after reconditioning.

30 It was further stated that " the boiler was a constant source of annoyance, steam having to be kept up at sea for stand-by purposes. Such large quantities of fuel oil were burned that the saving in maintenance and operation has been considerable and the installation of all-electric auxiliaries has effected an improvement all around. In addition, the boiler took up considerable floor space and crowded the engine room."

[1] Loc. cit.

31 The following average fuel-consumption figures obtained from several round trips to Chile with and without the steam auxiliaries illustrate the saving in fuel when using electric auxiliaries. It should be borne in mind, that the deck machinery is not used for either loading or unloading cargo.

32 As the Ore Steamship Corporation which operates the *Cubore* has a number of steamers about the same size operating on exactly the same schedule and conditions, Table 7 gives an idea of the economy that is actually obtained.

TABLE 7 COMPARATIVE OIL CONSUMPTION OF ORE STEAMERS

	Type of engines		
Fuel consumption per day	Reciprocating	Turbine	Diesel
Total, barrels	226.0	215.1	84.17
Total, tons	34.3	32.66	12.05
Port averages, total, barrels	80	80	4

33 The motorship *Seekonk* converted by the Wm. Cramp & Sons Ship & Engine Bldg. Co. is fitted with all-electric auxiliaries.

Auxiliary Machinery

3 100-b.hp. Diesel-engine generators (65 kw.)
1 Donkey boiler for heating, fire smothering, and heater coils at fuel-oil suctions.

Deck Machinery

1 Hydroelectric steering gear
10 3-ton electric winches
1 Windlass, electrically driven
1 Capstan, electrically driven.

34 Comparing figures with results obtained from steam sister ships of the *Seekonk*, the data of Table 8 are available:

TABLE 8 COMPARISON OF OIL CONSUMPTION OF STEAM AND MOTOR SHIPS

	Steamer *Hog Island*	Motorship *Seekonk*
Fuel per day at sea, main and aux. engine, tons (av.)	29.14	7.416
Donkey boiler, tons (av.)	....	0.513
Fuel per day in port[1] (rivers and harbors, etc., inclusive) tons	9.00	1.22
In port alone, tons	....	0.60

[1] The consumption in port varies from 0.3 to 1 ton, depending on whether the winches and donkey boiler are in operation, with a mean of about 0.7 ton during winter and 0.5 during summer months.

35 The motorship *Ashbee* converted by the New York Shipbuilding Corporation is fitted with steam deck machinery (except the steering gear which is hydroelectric); that is, the existing deck winches, capstan, and windlass were retained. Table 9 gives a list of auxiliaries.

36 Comparing the data available, Table 10 is obtained.

37 The Sun Shipbuilding Company has converted the motorships *Bidwell*, *Miller County*, and the *Challenger*. The first two

TABLE 9 LIST OF EQUIPMENT — MOTORSHIP *ASHBEE*

Auxiliary Engines

Three Worthington 60-hp., two-cycle, airless-injection oil engines direct-connected to three 40-kw., 220-volt General Electric generators.

Auxiliary air compressors

One Worthington 10½ in. x 7½ in. x 3½ in. x 8 in. direct-connected to 100-hp. d. c. motor. Capacity 225 cu. ft. per min.
One Worthington 4 in. x 3½ in. x 3½ in. x 3 in. steam-driven compressor 10 cu. ft.

Pumps

Five Worthington centrifugal pumps for circulating water, ballast, fire, and sanitary service, all direct-connected to G. E. motors
One Worthington triplex-plunger crank-action gear-transmission from G. E. motor.
Five Blackmer rotary-gear transmission, from G. E. motor drive, 2 for lubricating oil, 2 for fuel oil and 1 for bilge pumping.

Steam Plant

One vertical fire-tube boiler with 1390 sq. ft. of heating surface.
One small Wheeler condenser.
One air pump.
One circulating pump.
One bilge pump.
One boiler-feedwater pump.

Deck Machinery

Ten Lidgerwood 8 in. x 8½ in. winches.
One hydroelectric steering gear (Hyde).
One Main Electric Co. steam anchor windlass.

Refrigeration

One Remington ice machine, G. E. motor-driven.

Lighting

Original lighting panel and steam-driven dynamo.
One G. E. balancing set to work in conjunction with series circuits in converting 220 volts to 110 volts for lighting.

Lubricating-Oil Filtering

One 20 gal. per hour capacity DeLaval purifier.
One 100 gal. per hour Richardson-Phenix filter.

Machine Shop

Lathe, drill press, grinder, and stone, motor driven with common lineshaft.

TABLE 10 COMPARISON OF OIL CONSUMPTION OF THE SHIP *ASHBEE* BEFORE AND AFTER CONVERSION

Fuel per day at sea	*Ashbee* as steamer	*Ashbee* as motorship
Main and auxiliary engine, tons (avg.)	24.6	6.92
Donkey boiler [1]	....	....
Fuel per day in port, tons (avg.)		
Donkey boiler alone	....	1.29
Auxiliary engines alone	....	0.14
Total in port	7.6	1.43

[1] At sea only the main engine-exhaust boiler is used.

TABLE 11 FUEL CONSUMPTION OF THREE CONVERTED MOTORSHIPS

	Bidwell	*Miller County*	*Challenger*
Fuel per day at sea, main and aux., total tons	9.3	10.3	12.0
Fuel per day in port, tons	3.0	3.0	1.33

retained all their steam auxiliaries, whereas on the *Challenger* electric auxiliaries were fitted. Fuel consumptions of these ships as motorships are given in Table 11. No data are available of these ships as steamers.

38 Table 12 is a summary of the cases cited.

TABLE 12 COMPARISON OF SIX MOTORSHIPS

	Cubore			*Seekonk*		*Ashbee*				
	Average results of steamers practically same size	As motorship		Average results of steamer sister ships	As motorship	As steamer	As motorship	*Bidwell*	*Miller County*	*Challenger*
		With steam aux.	With Diesel elec. aux.							
Length, overall	.....	.....	468′-2⅜″	.....	401′-0 7/16″	.....	346′-6″	.....	.....	.....
Length, b. p.	.....	.....	450′-7″	.....	390′-0″	.....	334′-6″	430′-0″	430′-0″	410′-0″
Breadth, molded	.....	.....	57′-0″	.....	54′-0″	.....	48′-0″	59′-0″	59′-0″	56′-0″
Depth, to upper deck	.....	.....	.....	.....	32′-0″	.....	27′-6″	.....	.....	.....
Depth, molded	.....	.....	37′-0″	.....	27′-6″	.....	24′-9″	33′-3″	33′-3″	38′-0″
Draft, loaded	.....	.....	27′-6 3/16″	.....	24′-5 5/16″	.....	22′-5½″	.....	.....	30′-6″
Displacement, loaded, tons	.....	.....	15,992	.....	11,400	.....	8,125	14,760	14,760	15,930
Deadweight tonnage	.....	.....	11,000	.....	7,754	.....	5,532	10,260	10,260	11,620
Speed, knots	.....	.....	10	.....	10.25	.....	10	12.5	12.5	13.85
Brake horsepower	.....	.....	2,400	.....	1,770	.....	1,500	2,500	2,500	2,500
Fuel per day at sea, main and aux. total, tons	33.48	.....	12.05	29.14	7.416	24.6	6.92	9.3	10.3	12.0
Main eng. only, tons	.....	.....	11.478	.....	.....	.....	.....	.....	.....	.....
Aux. only, tons	.....	0.502	0.572	.....	.....	.....	.....	.....	.....	.....
Donkey boiler only, tons	.....	1.58	.....	.....	0.513	.....	.....	.....	.....	.....
Fuel per day in rivers, harbors, etc., tons	.....	.....	.....	22	5.93	.....	.....	.....	.....	.....
Fuel per day in port, total, tons	11.43	.....	0.57	8	0.6	.....	1.43	3	3	1.33
Aux. eng. only, tons	.....	0.47	0.062	.....	.....	.....	0.14	.....	.....	.....
Donkey boiler only, tons	.....	.....	.....	.....	.....	.....	1.29	.....	.....	.....

39 From the foregoing, the electrical auxiliary is evidently more economical than the steam, but how much of this saving, if any, is due to the use of electric deck machinery is a question that only can be solved for each particular case.

Comparison of Steam and Electric Winches

40 The advantages of the steam winch are as follows:

1 It is flexible and easy to repair
2 It will stop when the strain is too great, but the steam being just behind the valve, it is ready to pick up the load when the strain is relieved
3 The first cost is low.

The disadvantages of the steam winch are the following:

1 It requires a large boiler with all its auxiliaries; condenser, pumps, etc.
2 Large feedwater capacity must provide for the loss by condensation, leaks, etc., and
3 Long pipes decrease the overall efficiency.

The added apparatus means higher upkeep and repair expenses, and additional spares to be carried. The larger deadweight carried reduces the net cargo capacity and thereby the yearly revenue.

41 The comparison of the hypothetical ships made below is based on the following assumptions:

1 Both types of winches perform the same duty
2 Stevedoring charges are the same for both types
3 The period of time for loading and discharging is the same in both cases
4 The personnel is the same in both cases
5 The steering gear is hydroelectric in both cases.

The outstanding features of the two types of deck machinery are the following:

Ship with Steam Winches

1 Extra deadweight carried, or less net cargo carried
2 Fuel oil, lubricating oil, feedwater, etc., consumption larger in port
3 Lower initial cost.

Ship with Electric Winches

4 Greater net cargo carried
5 Less expense for power while in port
6 Higher initial cost.

42 The difference between 6 and 3 should, in the lifetime of the ship (or less), compensate or outweigh the other differences due to higher cost of handling a ton of cargo in port, as it is supposed that both ships at sea will be on equal basis.

43 It is understood, however, that due consideration to percentage of depreciation, insurance, and profit must be given to

each case and as the motorship of today is fully as reliable as the steamship, it is only fair to figure both on the same basis. This is generally 5 per cent per annum for depreciation, 5 per cent per annum for insurance, 8 per cent per annum (min.) profit on capital invested, giving 18 per cent per annum total.

Waste-Heat Boilers

44 No other part, except deck winches, in a marine oil-engine installation has been so misjudged and miscalculated as the waste-heat boiler. Reliable information on the subject is rather scarce. Several early installations of waste-heat boilers were made, but until recently the impression has been given that such boilers are a failure.

45 Although the exhaust gases carry away about 30 per cent of the total heat of combustion, the amount of heat that can be extracted and used for steam production is very much less, especially in two-stroke engines where the exhaust gas is diluted with scavenging air and has a lower mean temperature. Size and length of manifold, as well as location of waste-heat boiler, are other factors influencing the efficiency.

46 The gases generally leave the cylinder at a pressure of from 20 to 40 lb. per sq. in. and a temperature of from 600 to 1000 deg. fahr. By passing the exhaust gases from gas engines through a waste-heat boiler, it is possible in every-day practice to obtain an evaporation of slightly more than 2 lb. of water from and at 212 deg. fahr. per b.hp-hr. of the engine. In Diesel engines the thermal efficiency is higher than in gas engines and the temperature of the gas lower, which means that the output of steam available from the exhaust heat will be less per b.hp-hr. In actual practice on large engines, 0.8 to 1.2 lb. with a mean of 1 lb. of water from and at 212 deg. fahr., per b.hp-hr. has been evaporated at an exhaust temperature of around 600 deg. fahr.

47 Waste-heat boilers are generally fitted as a separate boiler unit, or, where steam boilers are retained for steam auxiliaries, the main boilers themselves are provided with an arrangement for cutting in and out the fuel-oil burners, and have proper bypass valves, generally of cast iron, with valves of the "flapper" or butterfly type of the same material.

48 Having manifolds, pipes, and valves all of cast iron, which of course is the ideal material, the weight (and thereby the price) is increased considerably. The life of steel pipe may not be the same as that of cast-iron pipe, but the price difference, weight, and installation might offset this to a certain extent.

49 The amount of steam in lb. per b.hp-hr. can be determined as follows:

Let T_1 = temperature of exhaust gas, deg. fahr.
T_2 = temperature of water or steam, deg. fahr.

t = temperature of gas leaving boiler over and above the temperature of hot water in boiler, deg. fahr.
H_1 = percentage of heat utilized in the boiler of the total carried by exhaust gas
H_2 = total heat developed per b.hp-hr., B.t.u.
H_3 = total heat which can be used for raising steam, B.t.u.
Q = calorific value of fuel, B.t.u. per lb.
g = fuel consumption, lb. per b.hp-hr.
h = ratio of total heat carried by exhaust gas and that of total heat of fuel burned
h_1 = total heat required to evaporate 1 lb. of water of the initial temperature T_3 deg. fahr. and the pressure p lb. per sq. in.
e = total amount of heat required to evaporate 1 lb. of water of the initial temperature $t_1 = 32$ deg. fahr.
f = heat contained in the water of temperature T_3, B.t.u. per lb.
η = efficiency of boiler, per cent
w = amount of steam generated, lb. per b.hp-hr.

Roughly

$$H_1 = \frac{T_1-(T_2+t)}{T_1}\,\eta \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

$$h_1 = e-f$$
$$H_2 = Q\times g$$
$$H_3 = H_2\times h\times H_1$$

or substituting,

$$H_3 = \frac{Qgh[T_1-(T_2+t)]}{T_1}\,\eta \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [2]$$

Thus the amount of steam per b.hp-hr. is

$$w = \frac{Qgh[T_1-(T_2+t)]}{h_1T_1}\,\eta$$

or

$$w = \frac{Qgh[T_1-(T_2+t)]}{(e-f)T_1}\,\eta \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [3]$$

50 For practical purposes h is generally about 30 per cent or 0.30, and g can be taken as 0.43 lb. per b.hp-hr. The boiler efficiency η can be taken as 90 to 95 per cent as it represents really only radiation and small losses, as all other factors affecting the steam production in the waste-heat boiler are already taken into account.

51 From Formula [3] it is apparent that a lower pressure in the boiler is advantageous, and the higher temperature of the exhaust gas essential. Thus the four-stroke engine is more suitable for working with a waste-heat boiler than a two-stroke one, which has lower temperature of the exhaust gases. The type of vessel,

machinery installed, service intended for, routes, etc., are factors to be considered in the installation of a waste-heat boiler as well as the design and construction of the boiler itself.

Determination of Costs

52 The number of days n for one trip is

$$n = \frac{L}{24V}$$

where L = length of trip in nautical miles
V = speed of ship in knots.

The number of single trips per year N is

$$N = \frac{365-d}{\frac{L}{24V}} = \frac{(365-d)24V}{L}$$

in which d = days in port or otherwise not sailing; or

$$N = \frac{365}{\alpha \frac{L}{24V} + \beta \frac{2t}{r}}$$

in which α and β are coefficients to take account of bad weather and delays in port, respectively, and

$\frac{2t}{r}$ = days in port, loading and discharging

t = net cargo, deadweight per trip, tons
r = rate of loading and unloading, tons per day.

$$\text{Fuel, stores, etc., required at sea} = \frac{L}{24V}\ (F+w+O+S)$$

in which F, w, O, and S refer to fuel, water, oil, and stores, respectively, at sea, in tons per day.

$$\text{Fuel, stores, etc., required in port} = \frac{2t}{r}\ (f+w_p+O_p+S_p)$$

in which f, w_p, O_p, and S_p refer to fuel, water, oil, and stores, respectively, in port, in tons per day. Hence T, the gross cargo deadweight in tons per trip, is

$$T = \left[\alpha \frac{L}{24V}\ (F+w+O+S) + \beta \frac{2t}{r}\ (f+w_p+O_p+S_p) + t\right]$$

and

$$t = \frac{T - \alpha \frac{L}{24V}\ (F+w+O+S)}{\beta \frac{2}{\ }\ (f+w_p+O_p+S_p)+1}$$

which can be written as

$$t = \frac{T_0 - \alpha F_0}{\beta f_0} \text{ or } \frac{T_0 - F_0}{f_0}$$

where T_0, F_0, and f_0 refer to the total, at-sea, and in-port tonnages, respectively. For maximum t or maximum cargo carried F_0 and f_0 must be minimum.

53 The freight revenue B is therefore

$$\begin{aligned} B &= N \times t \times R \\ &= \frac{365 - d}{L} \times t \times R \\ &= \frac{24V}{(365 - d) \times 24 \times R} \times V \times t \end{aligned}$$

in which R = freight rate in dollars per ton. Then, if L and d are constant, putting

$$\frac{(365 - d) \times 24 \times R}{L} = k_1$$

we have

$$B = k_1 \times V \times t$$

Net cargo weight can also be expressed by

$$t = \Delta - (W_1 + W_2 + W_3 + W_4)$$

in which Δ = displacement in tons, and W_1, W_2, etc., are the weights in tons of the hull, running gear, machinery, fuel, etc., Then writing

$$W_1 + W_2 + W_3 + W_4 = \Sigma W$$

we have

$$t = \Delta - \Sigma W$$

and

$$B = k_1 \times V \times (\Delta - \Sigma W)$$

54 One expense item E is approximately constant relative to the effectivity of the ship, and can be expressed as a function of the freight revenue, or

$$\begin{aligned} E &= \delta \times B \\ &= \delta \times N \times t \times R \end{aligned}$$

in which E = expenses of crew, maintenance, insurance, repairs, etc., if

$$\begin{aligned} \delta \times R &= k_2 \\ E &= k_2 \times N \times t \end{aligned}$$

55 Another item of expense E' depends on the number of trips and the cost of oil, water, port charges, etc.,

$$E' = N \left[\alpha \times n(FC_1 + wC_2 + OC_3) + \beta \frac{2t}{r} (fC_1 + w_p C_2 + O_p C_3) + ti \right]$$

in which C_1, C_2, and C_3 are the costs of fuel, feedwater, and lubricating oil in dollars per ton, and $i =$ cost of loading and discharging in dollars per ton.

$$N = \frac{(365-d)\times 24}{L} \times V$$

Now putting

$$\frac{(365-d)\times 24}{L}\left[\alpha n(FC_1+wC_2+OC_3)+\beta\ \frac{2t}{r}\ (fC_1+w_pC_2+O_pC_3)\right] = k_3\times W_4$$

we have

$$E' = k_3\times W_4\times V+N\cdot t\cdot i$$

Total expenses can thus be expressed by

$$\begin{aligned} D &= E+E' \\ &= k_2\times N\times t+k_3\times W_4\times V+N\cdot t\cdot i \\ &= N\times t(k_2+i)+k_3\times W_4\times V \end{aligned}$$

56 The depreciation a is directly proportional to the initial cost A and inversely proportional to the lifetime J. Therefore

$$a = \frac{A}{J}$$

and if J is a constant

$$a = k_4\times A$$

The initial cost is composed of cost of hull, running gear, equipment, outfit, etc., and machinery. We have then,

$$a = k_4(p_1W_1+p_2W_2+p_3W_3)$$

57 If we call $\Sigma k_1 = K_1$, $\Sigma k_2 = K_2$, $\Sigma k_3 = K_3$, and $\Sigma k_4 = K_4$, we have

$$\begin{aligned} \Sigma P &= K_1Vt-K_2Nt-K_3VW_4-K_4(p_1W_1+p_2W_2+p_3W_3) \\ &= V(K_1t-K_3W_4)-K_2Nt-K_4(p_1W_1+p_2W_2+p_3W_3) \end{aligned}$$

which again shows that the yearly income depends on

1 Cargo capacity t
2 Speed of vessel V
3 Weights of the different groups W.

For ΣP to be maximum $\partial\Sigma P$ must be equal to zero. For the special case that K_1 is equal to K_3 and the remaining part of the equation to the right remains constant, we have maximum ΣP expressed by

$$t-W_4$$

58 For the ship equipped with the steam winch we have

$$\begin{aligned} \Sigma P_s &= t_s-W_{4s}-t_s\times i_s \\ &= t_s(1-i_s)-W_{4s} \end{aligned}$$

and for the ship equipped with the electrical winch

$$\Sigma P_e = t_e - W_{4e} - t_e \times i_e$$
$$= t_e(1-i_e) - W_{4e}$$

It is evident that for maximum P_s or P_e, i_s or i_e must be minimum, as well as W_{4s} and W_{4e}, and that the pro and con hinge on the two factors i_s and i_e. Now the difference in cost of the two ships is

$$U = A_e - A_s$$

in which A_e and A_s are the costs of the electrically and steam-equipped ships, respectively, and this, considered as a sinking fund, must in years balance the cheaper operating expense in favor of i_e.

59 The annual payment Y which will amount with accumulated interest to the given sum U at the end of x years is equal to

$$Y = U \times y$$

where

$$y = \frac{0.01r}{[(1+0.01r)^x - 1]}$$

with r = interest, per cent per annum. Then we have

$$U \times y = N(i_s - i_e)$$

or

$$U \times \frac{0.01r}{[(1+0.01r)^x - 1]} = N(i_s - i_e)$$

and

$$\frac{U \times 0.01r}{N(i_s - i_e)} + 1 = (1+0.01r)^x$$

of which x can be found when i_s and i_e are known factors. The interest, r, is generally predetermined.

60 From data appearing in the press and founded upon experience and records collected (see also Table 12), the ratio between fuel consumed in port by steam auxiliaries and fuel consumed by electric auxiliaries is about equal to

$$\frac{f_s}{f_e} = \frac{10}{1}$$

and if everything else is equal or constant this gives a good index of the items i_s and i_e. Likewise for most motorships there is a somewhat constant ratio between fuel oil and lubricating oil consumed, as follows:

$$\frac{F}{O} = \frac{200}{1}$$

Feedwater consumed has a relation to fuel oil burned, and for practical purposes can be written

$$\frac{F}{\text{steam or water}} = \frac{12}{1}$$

Compressors and Air Storage

61 Compressed air as a medium for injecting and atomizing oil for the cylinder of a Diesel engine is used to a wide extent today just as it was the recognized practice in earlier types of engines using liquid fuel.

62 When the Diesel engine was introduced for marine work and large units were made reversible, in view of the handicaps under which a vessel labors at sea the capacity of the air compressor was made rather large, compared to that for a stationary engine of the same power. With the increasing knowledge of design, better castings and workmanship, etc., the last ten years have seen wonderful improvement in the reliability of the air compressors for marine-type Diesel engines.

63 To a certain type and size of ship there always corresponds a certain number and size of auxiliaries, and to minimize these without impairing or sacrificing the overall reliability and reserve service should be the chief aim of both engineers and manufacturers, although they are to a certain extent handicapped by the rules and regulations of the different classification societies.

64 Air compressors for injecting and atomizing as well as for starting purposes are no doubt at the mercy of the designing engineer, in regard to capacity and pressure for a given size and type of engine, and as an extra safeguard they are increased rather than decreased in most cases. In earlier Diesel engines an allowance was made for air-compressor capacity of approximately 0.5 cu. ft. per min. per b.hp., a figure now almost cut in two for modern engines.

65 The efficiency of some of those early compressors left much to be desired. Two-stage types were more common than three-stage types, and excessive low-pressure cylinder clearance, defective valves, bad material and workmanship, too great compression ratio per stage, etc., were all factors that resulted in a volumetric efficiency of around 60 per cent. The present three-stage air compressor fitted for Diesel-engine work has a volumetric efficiency of at least 80 per cent, and in some cases it is as high as 90 per cent.

66 To base the size of air compressors for marine work on the cu. ft. of free air per min. per b.hp. or on compressor- to cylinder-volume ratio is possible only where experience is the judge; but that the ratio should vary through so large a range, as shown in

Table 13, is hard to explain. It will be noted that the range is from 1: 24.40 (4.1 per cent) to 1: 12.375 (8.1 per cent) or double. A number of textbooks, magazines, etc., give a range varying from 1: 12 to 1: 24, where the small value is to be taken for large engines and the large value for small engines, also certain values in proportion to the number of cylinders, as follows: If

TABLE 13 RELATION OF SIZE OF AIR COMPRESSOR TO SIZE OF ENGINE

Engine No.	B.hp.	Ratio comp. vol. to cyl. vol.	Cu. ft. free air per hp. per min.	Remarks
1	900	1: 12.375	0.412	Regulation by unloader on L. P. stage
2	1500	1: 24.40	0.210	Regulation L. P. suction
3	1800	1: 21.00	0.260	" H. P. "
4	1500	1: 20.00	0.266	" L. P. "
*5	2000	1: 18.50	0.250	" L. P. "
**6	3000	1: 23.75	0.176	" L. P. "
***7	600	1: 13.65	0.290	" L. P. "

Compressors based on 80 per cent volumetric efficiency.

* Air compressor separately driven, with capacity for two engines. Capacity for one engine given in table, others directly attached to engine.

** German submarine high-speed engine.

*** 2-cycle engine, capacity of air compressors given is reduced to 4-cycle basis for comparison.

In all cases except 5 the air compressors are supposed to serve two engines if one compressor becomes inoperative.

value A equals the compressor- to cylinder-volume ratio and is taken as 1: 14 and based on a one-cylinder engine then, with the increase in number of cylinders, the value of A changes as follows:

Number of cylinders	1	2	3	4	5	6	8
Value of A	1: 14	1: 16	1: 18	1: 20	1: 22	1: 24	1: 26

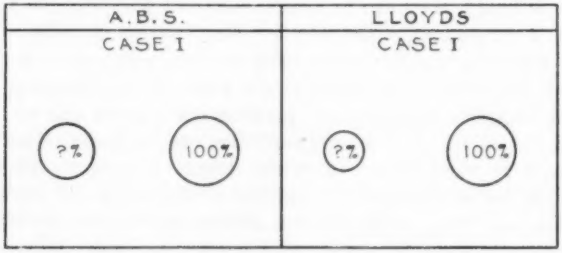

FIG. 2 DIAGRAM SHOWING RELATION OF AUXILIARY AIR COMPRESSOR TO MAIN AIR COMPRESSOR ACCORDING TO AMERICAN BUREAU OF SHIPPING AND LLOYD'S RULES, SINGLE-SCREW VESSELS

67 Table 13 gives figures taken from actual installations and a ratio of 1: 20, or 5 per cent, as an average figure, equivalent to 0.266 cu. ft. per b.hp. per min. That this is larger than necessary is borne out by the fact that trials on a 1500-b.hp. engine showed an actual consumption of 0.1 cu. ft. per b.hp. per min.

68 The capacity of both air compressors and storage tanks also affects (for a certain engine or given installation) weight, space requirements, and cost of production. The ideal capacity of course is the one with minimum capacity and still large enough for emergency purposes, or at least of such capacity that will safely bring the ship home, even at reduced speed.

Comparison of Shipping Rules

69 The rules of the classification societies are given below:

American Bureau of Shipping	*Lloyd's*
The number and capacity of the main air compressors should be ample to supply the air requirements of the main engines for a range of from 30 per cent to the full normal rated shaft power.	In single-screw vessels, an auxiliary air compressor is to be provided of sufficient power to enable the main engine to be kept continuously at work, when the main compressor is out of action.
Single-screw ships having main engines with attached or one independent main compressor are to have in addition one or more independent auxiliary compressors of ample capacity to supply the air requirements of the main engine at half the normal rated shaft horsepower.	If the maneuvering gear is arranged so that the engines can be kept continuously at work with some of the cylinders out of action the auxiliary compressor need only be of sufficient power to enable engines to be kept at work under these conditions.
Twin-screw ships having main engines with attached or independent main compressors are to have in addition one or more independent auxiliary compressors of ample capacity to supply the air requirements of one main engine at half its normal rated shaft horsepower. Where each independent main compressor has sufficient capacity to furnish the required air for the normal rated shaft horsepower of one engine and in addition for half the normal rated shaft horsepower of the other engine an auxiliary compressor will not be required.	In the twin-screw vessels, in which two sets of compressors are fitted, the auxiliary compressor is to be of such size as to enable it to take the place of either of the main compressors. If in such engines each main compressor is sufficiently large to supply both engines, a smaller compressor will be sufficient.
Twin-screw ships having main engines with one independent main compressor for supplying both engines are to have in addition one or more independent auxiliary compressors of ample capacity to supply the air requirements for the normal rated shaft horsepower of one engine or for half the normal rated horsepower of both engines.[1]	

[1] In addition to above, both societies require a small auxiliary injection air compressor, entirely independently driven as an extreme emergency. Therefore, as this compressor is only a very small outfit, only the large auxiliary and main compressors are considered.

70 In determining the capacity of the auxiliary compressor due consideration should be given the air requirements for maneuvering the vessel.

71 Better to illustrate the different requirements of both societies a graphic representation has been made, as shown on Figs. 2 and 3.

Single-Screw Vessels. American Bureau of Shipping rules clearly state the size or capacity, whereas Lloyd's rules leave this to the

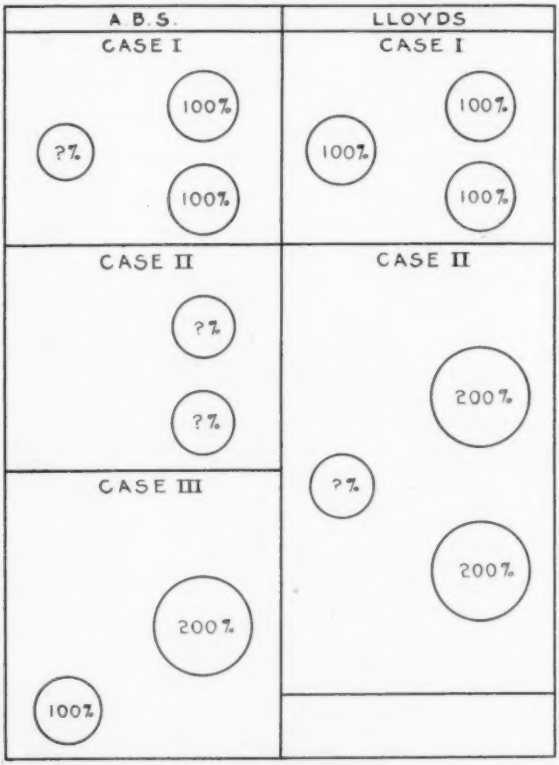

FIG. 3 DIAGRAM SHOWING RELATION OF AUXILIARY AIR COMPRESSOR TO MAIN AIR COMPRESSOR ACCORDING TO AMERICAN BUREAU OF SHIPPING AND LLOYD'S RULES, TWIN-SCREW VESSELS

judgment of the engine builder, and dependent upon the flexibility of the type of engine. The second paragraph in Lloyd's rules is merely a repetition of the first, as engines can be slowed down, and horsepower reduced or a few cylinders cut out, provision for this being made in most cases on marine engines, rendering a reduction in the propulsive power. But here, as before, the maximum reduction is limited by the flexibility of the engine.

Twin-Screw Vessels. American Bureau of Shipping rules again clearly define the capacity of the air compressors, whereas Lloyd's, in the last paragraph, leaves it to the designer or builder, to choose the capacity.

72 Horsepower, speed of vessel, and air-injection pressure and consumption are so interrelated that unless a definite rule for minimum speed of a vessel under adverse circumstances or distress is laid down, it is rather difficult to reach a happy medium or minimum figure in which all emergencies are met and still remain on the safe side. Much is dependent on having reliable auxiliaries of proper capacity, as lay-ups, breakdowns, etc. are always ultimately measured in terms of dollars, or loss to the owner. In other words, capacity is a function of (1) speed of vessel (function of time), (2) money (function of speed).

73 Before formulating the capacities of the air compressors based on experience and data derived from known installations in service, it is well to look into the fundamental principles involving the use of air for injection purposes.

74 The injection air pressure required depends on the type of fuel valve, the viscosity of the fuel used, the shape of combustion chamber, and the load and the speed of a given engine.

75 With larger cylinder diameters a greater injection pressure is required to give the fuel spray sufficient distribution and penetration throughout the combustion chamber. High-speed engines require air at a higher pressure than do slower-speed engines to give the fuel spray the required speed of entry, as a time element enters here.

76 The velocity of the fuel-air mixture entering the cylinder varies very nearly as the square root of the injection pressure, and the quantity of injection air consumed varies directly as the velocity and area of fuel-valve opening.

77 A fuel of high viscosity requires a higher atomizing force or injection pressure, otherwise atomization will be poorer and slower. Preheating a fuel of high viscosity is therefore beneficial in order to reduce air-compressor size.

78 In the majority of variable-speed Diesel engines the duration of fuel-valve opening is constant, while the period of oil injection varies according to the load.

79 At low loads, when all the oil has been injected, the air continues to enter the cylinder and uselessly expands, until the fuel valve closes. Besides a detrimental cooling effect, due to expansion, a large quantity of air is wasted.

80 At low r.p.m. and correspondingly slower speed, the volumetric efficiency of the air compressor increases, but the amount of air delivered decreases, due to fewer delivery strokes. From the

above it follows that best economy of air consumption is obtained when the fuel-valve lift is regulated to suit load and r.p.m.

81 If the horsepower vs. r.p.m. curve for a given engine is known as well as the most economical and beneficial injection pressure, it is easy to plot the most suitable injection pressure for partial loads.

82 Fig. 4 gives some interesting curves representing injection pressures for a given percentage load in horsepower.

Curve A is for a single-cylinder, 30-b.hp., stationary engine of 12-in. bore, constant speed.

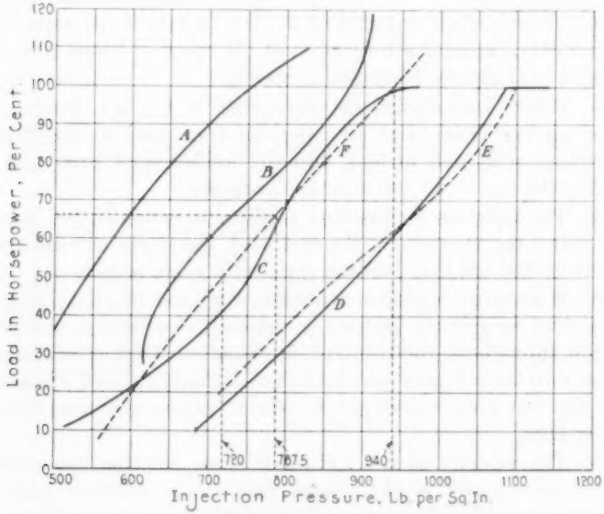

FIG. 4 RELATION OF INJECTION-AIR PRESSURE TO PERCENTAGE OF LOAD IN HORSEPOWER

Curve B is for a 3-cylinder, 200-b.hp., stationary engine of 17-in. bore, constant speed.

Curve C is for two 3-cylinder, 500-b.hp., stationary engines of 22-in. bore, constant speed.

Curve D is of a German submarine high-speed engine fitted with an automatic injection-pressure regulator.

Curve E is plotted for a large engine showing the most suitable pressure for a certain kind of fuel and at different loads.

Curve F is based on figures given by J. W. Southern in his recent publication, Notes and Sketches on Marine Diesel Engines.

83 The effect of variation of the injection-air pressure at different loads on the fuel consumption is illustrated in these curves, and it will be noted that there is a definite, most efficient injection pressure for each load. It will be noted that the curves A, F, and

D, and to a certain extent also curve E, are nearly parallel, giving a relation between injection pressure and load at maximum economy, or in other words most efficient combustion, whereas

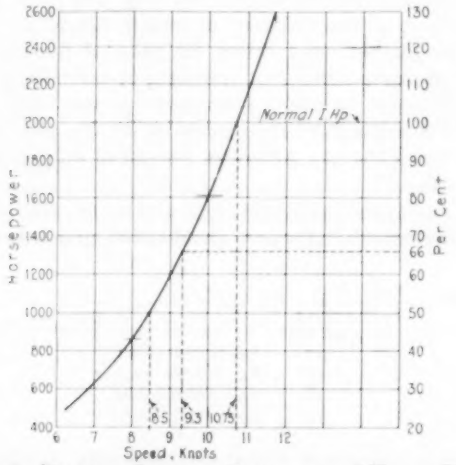

FIG. 5 DIAGRAM SHOWING RELATION OF I.HP. TO SPEED

curves B and C are left to the operating engineer and a "hit or miss" regulation, without any guide for efficient combustion. The great necessity of an automatic regulation of injection-air pressure

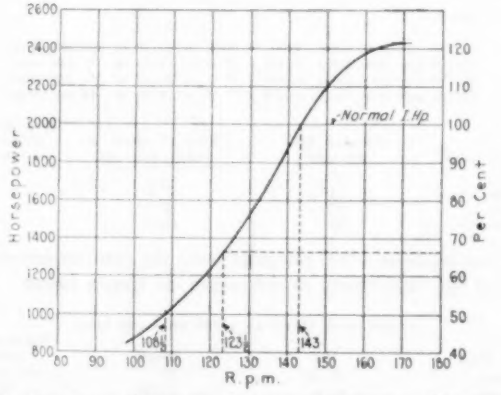

FIG. 6 DIAGRAM SHOWING RELATION BETWEEN HORSEPOWER AND R.P.M.

and fuel-valve lift for maximum economy can be seen. The curves will also reveal the increase in injection pressure for increasing size of cylinders, as well as for high-speed engines.

84 In Fig. 5 is shown a curve giving the relation between indicated horsepower and speed of a vessel under actual sea conditions. Examination of a large number of Diesel-engine test results reveals the fact that the difference between the i.hp. and the b.hp. remains nearly constant within ordinary operating ranges as the load is varied. The method used below for sake of illustration yields sufficiently accurate results for purposes of estimating in figuring a relation between air-compressor capacity and b.hp. load. The percentages of loads have been taken as 30, 50, and 66 per cent, the former being the new American Bureau of Shipping rules; the latter, the old.

85 We have now from Figs. 4, 5, and 6 the relations between injection-air pressure, horsepower and speed, and horsepower and r.p.m.

Velocities

Load, per cent	
100	$V_{100} = \sqrt{940} = 30.7$
66	$V_{66} = \sqrt{790} = 28.0$
50	$V_{50} = \sqrt{720} = 27.0$
30	$V_{30} = \sqrt{637} = 25.25$

86 The air-compressor output decreases with decreasing revolutions of main engine (attached compressor) due to a decrease in delivery strokes for a given period. The volumetric efficiency increases, however (if only a small percentage), with decreasing speed. Neglecting the latter, the capacity will be 100, 86.5, 76, and 52.5 per cent at 100, 66, 50 and 30 per cent load, respectively. Thus we have

At 100 per cent load $1.00 \times 30.7 = 30.70$ or 100 per cent
At 66 per cent load $0.865 \times 28.0 = 24.22$ or 79 per cent
At 50 per cent load $0.760 \times 27.0 = 20.52$ or 67 per cent
At 30 per cent load $0.525 \times 25.25 = 13.26$ or 43 per cent

Finally

Load, per cent	Drop in hp., per cent	Drop in speed of vessel, per cent	Drop in capacity of air compressor, per cent
100	..	....	..
66	34	13.5	21
50	50	20.9	33
30	70	36.5	57

Putting the loads on a 100 per cent basis the comparison or relation will reveal the uniformity or deviation, as shown below:

Compared with	Hp.	Speed	Air-compressor capacity
	30 per cent taken as a 100 per cent basis		
66 per cent	48.57	37	37.82
50 per cent	71.4	57.20	58.0
	50 per cent taken as basis		
66 per cent	68	64.6	63.6
30 per cent	140	170	172.5
	66 per cent taken as basis		
50 per cent	147	155	157
30 per cent	206	270	270.1

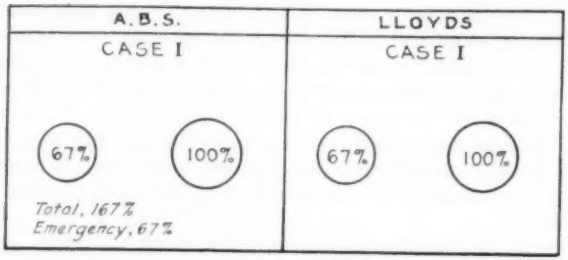

FIG. 7 MODIFICATION OF FIG. 2 TAKING 50 PER CENT LOAD AS A BASIS FOR AUXILIARY-AIR-COMPRESSOR CAPACITY, SINGLE-SCREW VESSELS

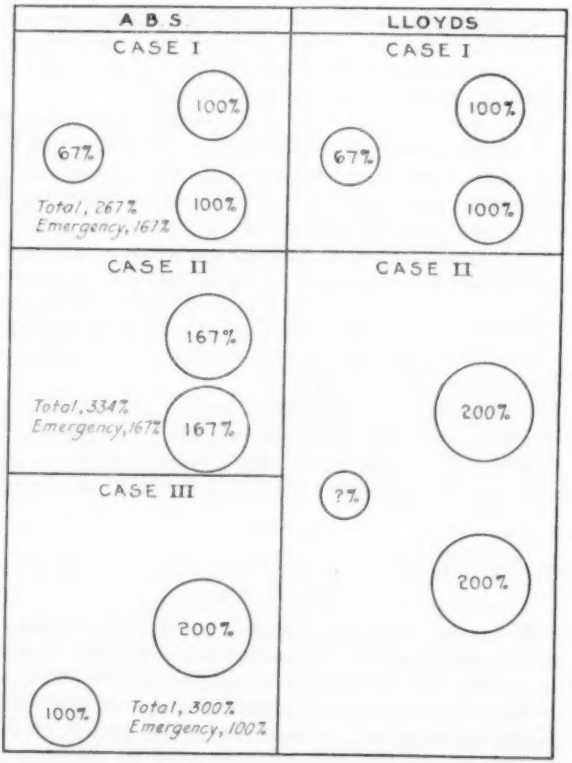

FIG. 8 MODIFICATION OF FIG. 3 TAKING 50 PER CENT LOAD AS A BASIS FOR AUXILIARY-AIR-COMPRESSOR CAPACITY, TWIN-SCREW VESSELS

87 Best uniformity in the figures (treated as index numbers), or minimum deviation, exist, as will be seen from the tabulation, on the 66 per cent and 50 per cent with the latter as a load basis.

88 Taking now 50 per cent load as a basis for the auxiliary air-compressor capacity, the Figs. 2 and 3, can be modified to give Figs. 7 and 8.

89 Single-screw vessels furnish no problems. For twin-screw vessels it will be noted that total capacity is 267, 334, and 300 per cent, with an emergency capacity of 167, 167, and 100 per cent, respectively.

90 The best case to employ is rather hard to decide, as first cost, flexibility, variation, and nature of breakdowns, arrangement of compressor and main engine, etc., are ruling factors in choosing the right layout. In most cases the tendency is to have main-engine air compressors directly attached and to have the auxiliary compressor independently driven Most direct-driven air compressors are of the slow-speed type, whereas the independent compressors are generally steam or electrically driven, with a higher r.p.m., resulting in smaller weight and floor space.

91 While there are pros and cons for high versus low r.p.m. for equal piston speeds, the higher-r.p.m. installation will have a higher number of reversals, which is really the determining factor in reciprocating machinery, as such factors as wear, balancing, lifetime, etc. are direct functions of the number of reversals.

92 Still it seems logical to have air compressors independently driven at high speed, thereby reducing weight and space occupied and at the same time reducing the length of the main engine, with a resulting shorter engine room in a fore-and-aft direction; but they will, on the other hand, increase the auxiliary power whether this be of the electric or steam type.

93 In all cases it is a matter of individual discrimination or space available that will determine the best and most economical arrangement and installation of engine-room machinery, whether single- or twin-screw vessels, whether a new ship or a converted one.

Capacity of Starting-Air Tanks and Auxiliary Starting-Air Compressor

94 The means universally adopted for starting marine Diesel engines of large power is compressed air, primarily because this medium is the most convenient to handle and the most flexible. The storage pressure employed is usually about one-third the pressure of the injection air. In considering the storage of this air, the question of capacity and pressure must be considered. The limits of pressure are between the lowest pressure at which the engine will start with a sufficient degree of certainty, and the pressure at which the relief valve is set (determined by the safe

working pressure of tank or tanks). Starting by means of high-pressure compressed air, say 1000 lb. per sq. in., reduces the air storage but increases the dangers connected with high-pressure systems, which require better, and therefore more costly, containers and fittings. To this end it is desirable, remembering conditions of safety, to reduce the starting-air pressure to a minimum. Present-day practice is to use a pressure of about 25 atmos. or 350 lb. per sq. in. The lowest pressure at which it is possible to obtain a sure starting of an engine is about 100 to 150 lb. per sq. in., depending on the size of cylinder bore.

95 It might be contended that the air storage should not be determined on a horsepower basis, but rather on a stroke, "filling-volume" basis. With due consideration to acceleration, normal speed in revolutions, and stroke-bore ratio, which are factors of importance, the former method of arriving at a minimum air storage is satisfactory, although a more accurate result is obtained by using the filling-volume basis. The advantage of a lower pressure system, as stated before, is the simplicity of the reservoirs, which may be of ordinary boiler or tank construction, meaning cheapness and increased safety.

96 Nine or ten years ago the following was predicted:

> In the future, the starting-air storage, whether high or low pressure, will undoubtedly be reduced when:
>
> 1 The confidence of experience is established
>
> 2 The regulation of the fuel is such that the chance of an engine failing to pick up on fuel immediately, due to any derangement of the fuel-injection-pump valves, or to any air in the fuel-pump passage or pipes, is minimized
>
> 3 Some system of preliminary warming up of the engine has been adopted.

The last clause is by far the most important of the three, as due to the slow compression at a low number of revolutions a considerable amount of heat passes from the cylinder walls into the jackets, keeping the cylinder comparatively cool. Further, the expansion of the starting air in the working cylinder from approximately the starting pressure, to the exhaust or atmospheric pressure, cools down the whole cylinder, and further aggravates the difficulty of starting. The heating of the starting air prior to its introduction into the cylinders is not practicable, but the assurance that the cylinders are warm before starting requires but a short time and little effort with steam, is beneficial in preventing them from being subjected to unreasonable heating, is easy of attainment, costs little, and would go far to reduce the starting-air storage to a minimum.

97 The above predictions are really realized in motorships of today. Still more discrepancy exists as to minimum starting-air storage and consequently in the capacity of the auxiliary-air maneuvering compressor, as manufacturers of engines vary in their specifications of air-storage reservoirs.

98 Table 14 gives some data of different types and makes of marine engines and the air storage specified by each manufacturer. In fact, the data are from motorships in actual service. It will be noted that great differences exist in the capacity of storage based on horsepower, as well as on cylinder volume. These relations are shown in Fig. 10.

99 A test carried out with engine 5 gave 20 starts from a tank of 250 cu. ft. capacity, without replenishing the air. The initial starting pressure was 352 lb. and the engine refused to start at a pressure lower than 100 lb., and at this pressure the time taken to start was nearly twice as much as with the initial pressure. Fig. 9 shows test results and relations between time, pressure, and the number of starts of this particular engine.

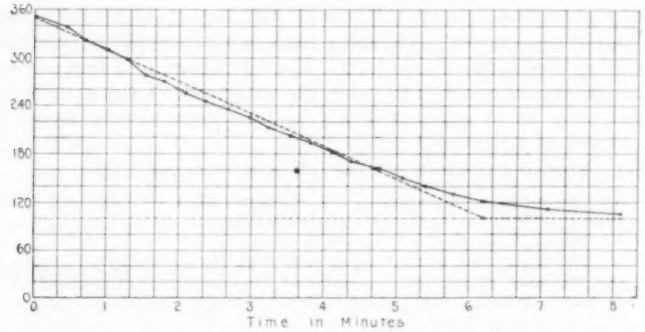

FIG. 9 DIAGRAM SHOWING RELATION BETWEEN PRESSURE, TIME, AND NUMBER OF STARTS ON ENGINE 5, TABLE 14

100 It will be noted from the table that a starting pressure of 25 atmos. is used in most marine installations, and can almost be considered as standard now. This leads to a general empirical formula giving the volume of the starting-air tanks in terms of the cylinder volume.

$$W_T = \frac{80VN}{P-14}$$

where

W_T = volume of starting reservoirs, cu. ft.
V = cylinder volume (piston area × stroke), cu. ft.
N = number of cylinders provided with starting valves
P = starting-air pressure, atmos.

Basing the formula on the actual filling volume,

$$V_1 = VL$$

where L is the part of stroke with starting valve open,

$$W_T = \frac{80V_1N}{L(P-14)}$$

A mean value of the coefficient L is generally 0.88 or 88 per cent for 4-cycle engines, and 0.72 or 72 per cent for 2-cycle engines.

101 The curve C in Fig. 10 shows the above formula plotted in connection with data given in Table 14. Curves A and B are plotted from Table 14, basing the starting-air storage on horsepower or cylinder volume (curve B) respectively. The curve C could have been plotted on a horsepower basis, but it will be seen from Table 14 that a great variety in r.p.m. and stroke-bore ratio exists. Take for example engines 5 and 2. A difference of 75 horsepower or 10 per cent exists in favor of No. 2, but this engine has a smaller cylinder volume, namely 0.25 cu. ft. or 3.5 per cent less than No. 5, and a difference in stroke-bore ratio of

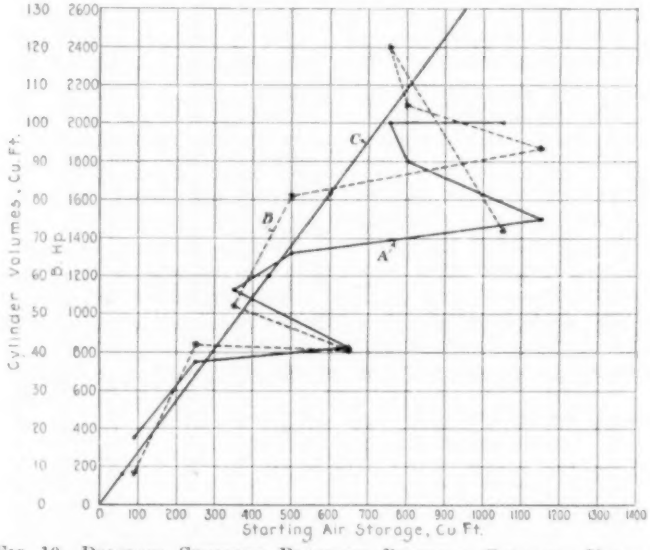

FIG. 10 DIAGRAM SHOWING RELATION BETWEEN CYLINDER VOLUME AND STARTING-AIR STORAGE VOLUME, AND BRAKE HORSEPOWER AND STARTING-AIR STORAGE VOLUME

0.305 or 21 per cent larger than No. 5. The r.p.m. are the same in both cases.

102 This also bears out the fact that the cylinder stroke-volume per horsepower varies widely. It is only natural then, to base the starting-air volume on a cylinder-volume basis. If the storage capacity, in the above case, had been determined on a horsepower basis, for engine 2 we would get a volume of 342 cu. ft. instead of 300, using the data from engine 5, of course, as a basis.

103 According to Lloyd's and American Bureau of Shipping rules only 12 starts, without replenishing, are required. Ample capacity is therefore found in the formula.

TABLE 14 AIR-STORAGE DATA OF 22 DIESEL-ENGINE INSTALLATIONS

1	Engine No.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22
2	No. of screws.....	2	1	2	1	2	1	2	1	1	1	1	1	2	1	2	2	2	2	2	2	2	2
3	B.hp., total	700	825	1000	1250	1500	1500	1650	2000	2000	2100	2250	2250	2250	2400	2400	2500	2640	3500	3500	3600	4800	3000
4	R.p.m.	265	135	....	....	135	110	....	96	100	....	....	....	140	....	....	.....	110	....	....	115	....	110
5	No. cylinders — each engine	6	6	....	....	6	6	....	8	6	....	....	....	6	....	....	....	6	....	....	6	....	6
6	Ratio, stroke÷bore ..	1.461	1.76	....	....	1.455	1.74	....	1.772	1.80	....	....	....	1.352	....	....	....	1.495	....	....	1.551	....	1.74
7	Cyl. stroke vol. in cu. ft.	1.382	6.75	....	....	7.00	15.56	....	15.00	12	....	....	....	8.66	....	....	....	13.50	....	....	17.45	....	15.56
8	Total stroke vol. in cu. ft.	8.292	40.50	....	....	42.0	93.36	....	120.0	72	....	....	....	52	....	....	....	81.0	....	....	104.7	....	93.36
9	Per cent filling.....	90	85	....	....	92	87.5	....	88	....	....	....	....	88	....	....	....	88	....	....	88	....	87.5
10	Actual filling — item 7×item 9.....	1.244	5.74	....	....	6.44	13.62	....	13.20	....	....	....	....	7.62	....	....	....	11.9	....	....	15.36	....	13.62
11	Total actual filling — item 8×item 9.....	7.464	34.44	....	....	38.64	81.72	....	105.6	....	....	....	....	45.72	....	....	....	71.28	....	....	92.16	....	81.72
12	Normal air pressure at starting valve, atmos. abs.	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25
13	Normal air pressure at starting tanks, atmos. abs.	25	25	25	35	25	25	60	25	25	75	40	48	25	25	25	75	25	25	60	25	25	25
14	Total vol. start. air tank, cu. ft.	180	650	215	210	500	1150	230	755	1050	150	220	200	700	500	1130	155	1000	1520	450	1600	1175	1500
15	Ratio to cyl. vol. based on 25 atmos. pressure	10.87	16.05	5.9	4.73	5.95	12.35	13.1	6.3	14.60	8.95	4.05	4.43	6.72	2.74	9.00	6.2	6.17	6.32	4.83	7.65	4.22	8.03
16	Items 13 & 14 reduced to free air vol., cu. ft.	4500	16250	5375	7350	12500	28750	13800	18875	26250	11250	8800	9600	17500	12500	28250	11625	25000	38000	27000	40000	29375	37500
17	Min. air press. for start'g, atmos. abs..	8	8	8	8	8	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10
18	Items 14 & 17 reduced to free air volume, cu. ft.	1440	5200	1720	1680	4000	11500	2300	7550	10500	1500	2200	2000	7000	5000	11300	1550	10000	15200	4500	16000	11750	15000
19	Actual free air volume available for starting, cu. ft.....	3060	11050	3655	5670	8500	17250	11500	11325	15750	9750	6600	7600	10500	7500	16950	10075	15000	12800	22500	24000	17625	22500
20	Cu. ft. capacity air tank per b.hp. at 25 atmos.	0.257	0.788	0.215	0.235	0.333	0.767	0.334	0.378	0.525	0.214	0.156	0.170	0.312	0.208	0.470	0.186	0.378	0.434	0.308	0.444	0.244	0.500
21	Total volume calculated according to formula, cu. ft. ...	125	300	....	....	625	685	....	875	525	....	....	....	760	....	....	....	1200	....	....	1550	....	1360

Auxiliary Air Compressor

104 The capacity of the auxiliary air compressor is more or less subject to variation and depends on whether the installation is for a single-screw or twin-screw ship, also on the capacity and pressure of the starting-air reservoirs, and the manner of driving them. Having more or less established the capacity of the latter, and the approximate capacity of the compressor from the rules of the classification societies, there remains a third factor to be taken into account, namely, the time taken to fill the starting-air reservoirs.

105 Investigating several motorships in service, it was found that the capacity of the auxiliary air compressor for starting and maneuvering was such that it was able to fill all the tanks or reservoirs in 90 min., in both single- and twin-screw vessels. It will, therefore, be found in many cases, that, when using the above time as standard, the auxiliary air-compressor capacity will be larger than the capacity called for by the classification rules.

106 Generally there is only one starting-air reservoir in operation at any one time, the filling of the other or others requiring only half or part time. Usually less time is taken, as only few starts are required, due to the reliability in starting nowadays. Besides the auxiliary air compressor which is replenishing the air used for maneuvering, the main-engine compressor in case it is directly attached is generally opened up wide, bypassing some of its air into the starting reservoirs. It is safe, however, to base the auxiliary compressor on the time allotment given above, and the capacity in cu. ft. of free air per min. is given by the following formula:

$$W_c = \frac{WP}{90}$$

where

W = volume of starting-air tank in cu. ft.
P = starting-air pressure, atmos.
W_c = compressor capacity in cu. ft. of free air per min.

107 The above, while open to criticism, is an effort to minimize the capacity of starting-air tanks and auxiliary compressors, though still on the safe side, and should therefore be of some value to engineers preparing Diesel-engine installations.

PART II

By A. B. Newell

108 As the fuel consumption per indicated horsepower of the steamship decreases there is an increase in the number of necessary auxiliaries. We may take as an example the older ships with Scotch boilers and triple-expansion engines and compare them

with a ship equipped with high-pressure, superheated-steam turbines. The difference is at once apparent. The tendency with motorships is the other way. Conservative methods demand simplified practice and additional auxiliaries would tend toward complication. The solutions of the opposite trend in the two different methods of ship propulsion are prompted, however, by the same object, i.e., reduced cost of operation.

109 Steamships must effect greater fuel economy in order to compete successfully with motorships on long voyages, for, as the result of long periods of continuous operation they must shut out cargo to make room for fuel. On the other hand, the motorship must reduce the capital charges to compete successfully in short hauls, where the question of shut-out cargo may be almost neglected. Time in port must either be charged to cargo handling or to profit and loss. To carry expensive equipment which is not needed, costs dearly.

110 In short, the steamship must reduce fuel consumption and the motorship must reduce the initial cost. In the first instance more auxiliary machinery seems necessary, while in the second, less seems essential. The figures involved may be summarized as follows:

Depreciation	5 per cent
Interest or profit on investment	8 per cent
Insurance	5 per cent
Repairs and maintenance	2 per cent,

making a total of 20 per cent per year on the difference in cost. This figure must not be overlooked when thinking of a 60 per cent saving in cost of fuel for the motorship. The 20 per cent continues whether the ship runs or not and represents a fixed amount, whereas the 60 per cent changes almost directly as the total fuel consumption changes upon the vessel entering or leaving port.

111 For a number of years, motorships were looked upon in America as experiments. When they were first tried, investors, feeling uncertain of the outcome, resorted to a number of foolish expedients hoping to reduce the first cost. The cheapest, and consequently poorest, auxiliaries were purchased. Sometimes second-hand steam machinery was used; old boilers, old pumps, old generators and condensers were gathered together and crammed into the engine room regardless of their adaptability or the space they occupied. Needless to say, grief resulted.

Engine-Room Auxiliary Machinery

112 The tendency now is to follow the more mature methods of European builders, with certain modifications which we may call Americanizations. We may not be original in many respects, but we have a certain individuality. Thus the engine-room auxiliary machinery of freight motorships of from 5000 to 10,000 tons dead-

weight capacity have been reduced almost to a standard, so nearly so at least that the following list may be taken as representative of the average.

PUMPS

	Service	No. of units	Type
1	Main cooling water	2	Centrifugal (generally)
2	Piston cooling water	1	Plunger
3	Lubricating oil	2	Rotary or plunger
4	Fuel-oil service	1 or 2	Rotary or plunger
5	Bilge and ballast	1	Plunger
6	Engine-room bilge	1 or 2	Plunger
7	Sanitary	1	Plunger or centrifugal
8	Fresh water	1	Plunger

Generating Sets. Generators driven by small oil engines, not less than two and seldom more than three, to supply current for the auxiliary machinery in the engine room, steering gear, deck winches, and anchor windlass, and for lighting the ship. The maximum output of these is sufficient for the maximum load, while the normal sea load may be taken care of with but one.

Fresh-Water Cooler for Piston-Cooling Service. Similar to a boiler feedwater heater or a steam condenser.

Compressors. One auxiliary compressor with a capacity slightly less than that of the main engines. One emergency compressor with a small capacity, sufficient to charge a starting bottle and put a generating set in operation.

Emergency Lighting Set. If steam is used for any purpose this may be steam driven, with sufficient capacity for lighting the ship and running the bilge pump, or the emergency compressor. If no steam is used a small gasoline or kerosene engine which may be started by hand, and which has a similar capacity, is employed.

Heating Plant. A small oil-fired boiler, sometimes so arranged that exhaust gases from the main engines will generate steam while at sea.

Water Traps and Feed Tanks. Where a closed heating system is used with no steam machinery employed, no condenser is needed.

Boiler Feed Pump. Generally electrically driven with a bypass to the fresh-water service pump giving a dual system.

Oil Filters or Centrifugal Purifiers. Various methods of oil purification or filtration have been employed. As in the case of other engine-room auxiliaries a standard method seems imminent, and is resolving itself into the use of centrifuges. Other methods are wasteful and dirty. The centrifuge removes impurities, leaving them in a semi-hard or plastic condition within the bowl. The filter or multiple strainer leaves the impurities in a heavy liquid state. The importance of a suitable method of reclaiming used lubricants is at once apparent when we realize that from 300 to 2000 gallons of lubricating oil may be in circulation and there is always the possibility of an inadvertent contamination with water, which would render the oil useless unless it could be reclaimed. Naturally other contaminating elements such as carbon, rust, and mill scale from the tanks will need to be removed from the oil from time to time.

Machine Shop. Generally well equipped with lathe, drill press, grinding wheels, and work bench, and sometimes a blacksmith outfit as well.

113 The foregoing list is the common arrangement where the ship is electrically equipped throughout. Several modifications may be made to suit different opinions. The ship may be heated

electrically and the small boiler done away with. If the auxiliary oil-engine generating sets use the air-injection principle of fuel atomization, oversize compressors may be substituted for the regular attached compressors and the surplus of air thus obtained would eliminate the need of the auxiliary compressor. Where steam winches are used a donkey boiler with condenser, air pump, feedwater pumps, and the regular oil-firing equipment would be added. In this event the generating sets might be made smaller, and the auxiliary compressor steam driven, since it is only used while maneuvering in or out of port. An alternative arrangement which is losing popularity is to have all of the auxiliaries enumerated (the generating sets may be reduced in number and power) driven with steam. This involves a sizable steam plant, materially increases the sea fuel consumption, and keeps the engine room uncomfortably hot in the tropics, but makes it comfortable in cold climates. This arrangement has never effected a saving in cost equal to the saving in fuel when electrical equipment was used throughout. It was formerly employed for the sake of dependability, a need which no longer exists, since up-to-date auxiliaries of the oil-engine electric type are fully as dependable as steam.

114 The fact that certain types of pumps have been almost universally accepted is worthy of some explanation, for this standard was not reached without cause. If steam is used the type of pumps will not be the same as given in the list. Naturally there is no need to make a departure, except in capacity, from standard steamship practice in such instances. What follows is with regard to electrically driven pumps.

115 The centrifugal pump is used for main circulating water because there is never any question of the supply of water, it requires practically no attention, repairs when needed are inexpensive, and the speed easily conforms to an efficient and economical speed of electrical machines. The head against which it pumps is relatively small.

116 For piston cooling the plunger pump is used with fresh-water circulation (if salt water is used the main circulating pump takes care of that as well) primarily to insure dependable service. Fresh water must be carried in tanks, and the ship in rolling will uncover the suction pipe if the tank is partly empty. With air in the centrifugal pump it ceases to pump, but the plunger pump works along. The same thing applies to bilge, ballast, and fresh-water pumping.

117 Oils are handled well with rotary pumps for two reasons: they do not lose their priming easily, and the oil keeps them well lubricated. They have lost favor for bilge pumping because water is a poor lubricant at best and the bilge strainers do not remove small particles of grit such as rust and mill scale which are ever present in bilges. These fine abrasives keep cutting the wear-

ing surfaces and repairs assume a place of importance. They may be used as ballast pumps but tend to give the same trouble, on a smaller scale, since certain abrasive sediment accumulates in these tanks also.

118 Common practice is to have an auxiliary compressor of the three-stage type similar to the main compressor. An alternative arrangement is to have a two-stage machine and an independent booster. With this latter the two-stage compressor pumps direct to the maneuvering tanks, raising the pressure to 350 lb. per sq. in., or less. The booster takes that air and lifts it to 1000 lb. or less.

119 The problem of auxiliaries for twin-screw installations does not differ greatly from that of the single-screw job. The proportionate amount of air storage is somewhat less in so far as maneuvering air is concerned, for a smaller engine will start with less air than a large one and the tendency in maneuvering is to use one engine about half of the time, particularly when the ship does not carry much weight or headway and is easily checked. However, a complete dual high-pressure-air system is required for each engine, which tends to balance the reduced amount of maneuvering air, as far as cost of installation is concerned.

120 A machine shop has been mentioned. This is more essential on the motorship than on the steamship. In fact tackles, cranes and substantial lifting gear are almost indispensable. Every provision must be made for doing routine repairs with a minimum of labor expenditure.

121 Although this paper is not touching upon the question of repairs, the last paragraph would be misleading unless this point were cleared up. The routine work on a motorship requires about the same amount of labor that such work on a steamship does. However, boiler work is eliminated, and this all applies to the engine room, particularly the main propelling units. The ship's engineers do this work because adequate means of having it done ashore have not been provided in all ports. It is still difficult to find mechanics in repair yards who can work intelligently on motors. The various parts to be handled, such as valves, cylinder heads, and pistons, are heavy. The engine-room staff is small. To solve this problem the best of gear has been found to be necessary.

122 Machine-shop work generally takes the form of facing valves and making small parts which may be broken or damaged during repair work. Not only at sea but in port the small shop pays for itself, for many of the small parts can be turned out aboard ship quicker than a messenger can go to the shop ashore and get them, if they have to be made in any event.

123 The trend is to reduce the number of auxiliaries with a view to reducing the cost, but nothing of quality may be sacrificed for the same purpose, for the main propelling unit is just as dependable as the auxiliaries, and no more.

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DISCUSSION

W. Mulheron.[1] Referring to Pars. 6 to 11, electrically driven auxiliary machinery has not only been developed to a point where it is, in every respect, as reliable in service as the best type of steam plant, but it is always operating at maximum efficiency, provided the units have been properly selected for the service. The efficiency of electrically driven machines is practically constant, whereas it requires constant maintenance and attention to keep up the efficiency of steam-operated auxiliaries. Where Diesel-engine-driven generators are used as the source of auxiliary power, their efficiency is also practically constant, whereas the steam consumption of steam pumps, winches, steering engines, generating sets, etc., is not only bad to begin with, but rapidly gets worse as time goes on.

The author has very properly pointed out that, although the electrical auxiliary machinery is inherently more expensive than equivalent steam machinery, the higher initial cost will be written off completely in a relatively short time, because of the remarkable economies obtainable with electric auxiliaries. Furthermore, as the direct-current system is almost universally used on shipboard, together with the fact that electrical equipment has been so standardized as to make repairs and replacements a relatively simple matter, a high degree of special knowledge is not required by the average seagoing engineer.

The leading manufacturers of electrical equipment build motors and control particularly for the service required, and experience has demonstrated the requirements to be met for marine service, the principal ones being:

a Insulation designed to withstand moisture and oil vapors in engine rooms of Diesel ships

[1] Mechanical Engineer, Marine Dept., General Electric Co., Schenectady, N. Y. Assoc-Mem. A.S.M.E.

b All fittings of non-corrodible material or with non-corrodible surfaces

c Self-oiling bearings not affected by rolling or pitching; 150 deg. tilt is accepted as satisfactory.

The development of electrical equipment for cargo winches has been particularly difficult, because of the abuse to which such equipment is subjected, as pointed out by the author in Par. 11. The development of electric-cargo-winch control, however, has reached the point where it is superior in every way to the steam winch.

In Pars. 40 and 41 the author has given a comparison of electric and steam winches. The writer cannot agree with him that steam winches are more flexible and easier to repair than electric winches. The electric winch is essentially of simpler construction than the steam winch, and, provided repair parts are available for both types of winches, it is as easy to effect a repair on an electric winch as on a steam winch. Furthermore, the experience of companies operating both types indicates more repairs and maintenance on steam than on electric winches. This question of repairs and maintenance of winches is very important in its relation to operating costs, as the breakdown of a winch, when handling cargo in a port, is exceedingly costly. The cost of repairing the machine itself is small as compared to the costs resulting from delay. That a ship is earning money only when she is at sea, is an axiom among ship operators, and therefore the whole development in electric-cargo-winch design has been not only to surpass the steam winch in speed of handling cargo, but to make it a far more reliable and economical piece of machinery.

In Par. 12, referring to voltage, the author states that 110 and 220 volts are in general use. The Marine Rules of the A.I.E.E., which are generally accepted today in this country for purposes of design, specify the voltage of generators to be 240 or 120, and the voltage of motors to be 115 or 230. A voltage of 230 is generally used for power purposes and 115 volts for lighting, the lower voltage for the lighting circuits being obtained by a motor-generator set or a balancer set.

There have been installations of three-wire generators, but this is not generally favored, partly because of the wide fluctuations in unbalanced load, and secondly on account of the expense.

A higher voltage is not favored for lighting systems, because of the greater durability of low-voltage lamps, and furthermore, greater stocks of 110-120-volt lamps are maintained throughout the world, and are available in foreign ports for replenishment, which is not the case with lamps for higher voltage. Difficulties with the lighting system, due to grounds, etc., are greater with 230 volts than with 115 volts. Aboard ship, it is surprising how

much trouble is experienced with grounds in wiring devices, due, of course, to dampness, vibration, etc.

In regard to Par. 15, the writer would like to point out that it is becoming the generally accepted practice to fit generators and large motors, whose physical dimensions approach those of generators, of the open type. Sometimes a sheet-metal cover is provided over such machines to protect them from dripping water, but otherwise they are entirely open. Small engine-room motors are generally of the semi-enclosed type, or enclosed ventilated type. In either case, openings are generally fitted with screens to prevent the entrance of rodents, which destroy the insulation. Motors for deck machinery, where exposed to the weather, are of the enclosed, waterproof type. As pointed out by the author, all motors and controllers mounted on the open decks must be not only waterproof but of very rugged construction to withstand the rough treatment to which they are subjected and to prevent damage from heavy seas.

The design of winch described in Pars. 17 and 18, and shown in Fig. 1, would involve very great difficulties. As the author points out, the oil engine would have to be large enough to handle all winches to which it might be connected at one time. As different winches would be operated at different speeds, and some lowering while others are hoisting, friction clutches would have to be used. Such clutches have never proved satisfactory, particularly when the winches are on the weather deck. Friction-drive winches have been popular with some of the coastwise steamship lines, but they are almost invariably located between decks, and, even so, their maintenance has been high.

A scheme somewhat similar to this was tried on some ships built during the war, where a constant-speed, alternating-current motor was connected by gears and clutches to two winches. This arrangement was not successful.

One of the great advantages which electric deck machinery has is the freedom from trouble experienced with steam deck machinery in winter. The steam and exhaust lines to deck machinery must, of necessity, be carried outside on the weather decks. If the freezing of the steam lines to winches with consequent bursting of pipes and cylinders is to be avoided, it is necessary to maintain steam pressure on all of these pipes in cold weather, and in severe weather it is the practice to keep the winches turning over slowly to avoid freezing. This, of course, involves very great steam losses, and, even with the best of care, pipes are burst and cylinders cracked.

In Par. 19 the author points out that the average loads handled by cargo winches vary between 700 and 1400 lb. Experience with many ships, and in many ports, indicates that in handling general cargo this is true, and the tendency in modern winch design is to base the design on handling average loads of from

1000 to 1500 lb. at the highest practical speed, in order to obtain the maximum tonnage rate for average conditions. For handling heavy loads there is provided either change-speed gearing, or a shift in the boom rigging from a single whip to a multiple-part fall, or a combination of both.

In Par. 22 the author points out the effect on steering of a careless and a skilful helmsman. The development of the well-known course recorder and gyropilot is due to this condition. The course recorder shows graphically the variations in course direction. This question is of far greater importance than the mere steam or current consumption of the steering engine, as it has long been realized that the speed of the ship is directly affected by the amount of helm necessary to keep the ship on a straight course. This has led to the development of automatic steering by gyroscopic apparatus, and recent investigation at the U. S. Experimental Model Basin indicates that it is entirely feasible to reduce the installed shaft horsepower of a given vessel and still maintain the average sea speed desired, when automatic steering is used.

In Pars. 69 to 107 the author has covered the question of air compressor capacity and starting-air tank capacity in a very able manner. He has neglected, however, to mention that with the Diesel-electric form of propulsion, where the Diesel-engine-driven generators are run at constant speed and are non-reversible, the necessity for maneuvering is entirely eliminated. The air-compressor capacity is required to be only that actually necessary for the operation of the engines, as there is always sufficient excess in the low-pressure stage to maintain the small starting-air tanks at the required pressure.

The author has also neglected to mention compressors which are required with two-cycle engines for scavenging purposes. Dr. Sanford A. Moss has covered this subject so thoroughly in his paper[1] that the writer merely wishes to call attention to this oversight.

JAMES A. SHEPARD.[2] In Part 1 the author has brought together many valuable data and makes individual deductions which, in the main, cannot be seriously questioned. Having submitted evidence pro and con, the lack of a conclusive verdict, or even of a prediction calculated to guide the inexpert, is a matter for regret.

The author's comments in Par. 3 relative to the "appalling inefficiency" of steam-driven auxiliaries are excellent. Exception may be taken, however, to his assignment of reasons for their continued use. Habit, precedent, timidity concerning new problems, and an insensible leaning toward a low initial cost, thus

[1] See *Mechanical Engineering*, vol. 47, No. 11a, Mid-Nov. 1925, p. 1075.

[2] Vice-President, Shepard Electric Crane & Hoist Co., Montour Falls, N. Y. Mem. A.S.M.E.

avoiding the necessity for explanations, and added responsibility, may also be prominent factors influencing, more or less, all who coöperate in the final selection of equipment, rather than actual merit.

The advent of entirely successful electrically driven deck machinery is comparatively recent. Hence, we may assume that results so far observed do not cover all advantages potential to electrically operated equipment. Also, we may assume that they may be more or less diluted by crudities in design, construction, and mode of application, so that with natural ultimate progress in the art much more favorable results than any so far observed may be expected from electrically driven auxiliaries.

An important difference in performance between steam and electric propulsion is partially noted by the author, viz.: that the former will stall when slightly overloaded. That the latter will continue to operate with overloads of several hundred per cent is a matter of the utmost importance. This inherent difference has rarely been suitably reflected in the construction of electric cargo winches. Hence, frames, gears, bearing, etc., follow the precedents previously established for steam winches. A record of equal reliability therefore falls below the possibilities of the electric winch, which, if properly designed, constructed, and applied, should develop far higher efficiency, which, unlike the steam winch, will be retained throughout its life. At the same time greater reliability and less in the way of care and repairs is to be expected.

The performance of the steam and electric winch is assumed by the author to be equal. That undoubtedly has been heretofore their relative status. Recent improvements in methods of control of the electric winch, providing very high lowering speeds for the empty load hook with less consumption of electric current than formerly, afford at once still greater relative efficiency, while performance is raised from mere equality to a point far exceeding that obtainable with the steam winch.

The average cost of loading and unloading cargo has been estimated at from 20 to 25 per cent of the direct disbursements for operating cargo ships. The proportion of time spent in port may exceed 25 per cent; hence, the efficiency with which cargo is loaded and discharged must bear a very intimate relation to total operating efficiency.

The mechanical efficiency of the cargo-handling equipment, while important, is but one of several factors which should be resolved into the one all-important consideration: net operating efficiency of the ship. In such a determination not only electric winches versus alternatives should have consideration, but the right kind, the right number, the right arrangement of electric winches versus alternatives should be weighed. Such a determination will rarely, if ever, leave room for doubt concerning the superiority of electric deck auxiliaries.

Mr. Newell refers to the foregoing consideration indirectly in Part II of the paper, although he does not recognize the advantages which may be realized by shortening the time in port through the use of more and better equipment for loading and discharging cargo. Such an oversight is excusable in view of the fact that we are only on the eve of developments, available through the use of electrically operated cargo-handling equipment, which will in any determination of overall efficiency leave little, if any, room for argument as to the superiority of electrically operated auxiliaries, whether the ship be propelled by steam or oil engines.

LESTER M. GOLDSMITH.[1] The writer has designed and placed in operation five Diesel-electric-propelled vessels. It may be of interest by way of comparison, therefore, to give a few figures on the auxiliary load of such vessels.

On the Diesel-electric-driven tanker *J. W. Van Dyke*, a vessel of 2200 shaft horsepower, it is interesting to note that only 2½ per cent of the total brake horsepower developed, and therefore only 2½ per cent of the fuel, is required for auxiliaries; whereas, quoting from the authors' paper, on a vessel of the same brake horsepower, with steam auxiliaries, approximately 22 per cent of the fuel is required for the auxiliaries, and with electrically driven auxiliaries, approximately 3 per cent, taking all of the foregoing as overall or average figures.

Assuming the power taken by the compressor as from 10 to 15 per cent of the total power of the main engine, we should add this amount to the power taken by other auxiliaries, which in the ships mentioned would bring the total percentage of power taken by auxiliaries to from 12 to 18 per cent. It therefore appears that on the *J. W. Van Dyke* we get an extra 9½ to 15½ per cent of the total power available for propulsion. In connection with these figures it should be noted that the *J. W. Van Dyke* engines are of the mechanical, or solid-injection, type, and therefore in comparing them with the air-injection type the injection compressor should rightly be considered as an auxiliary. Aside from the above the electrically propelled vessel requires no extra air for maneuvering as it is maneuvered electrically, and hence, when in restricted waters, has no greater auxiliary load than when at sea.

In addition, such a vessel has no auxiliary generator sets, as the main Diesel-electric generating sets also supply the auxiliaries, and at the same time there is no need for stand-by Diesel-electric sets for auxiliaries. This of course means a reduction of the capital cost, notwithstanding the other innumerable advantages. The motorship *J. W. Van Dyke*, being equipped with solid-injection engines, has no high-pressure air compressors for injection, and therefore requires air only for starting.

[1] Consulting Engineer, Atlantic Refining Co., Philadelphia, Pa. Assoc. Mem. A.S.M.E.

Relative to the formula, Par. 106, giving a method for computing the compressor capacity for starting air, it may be of interest to note that if we work out this formula, as applied to the *J. W. Van Dyke,* it would then read

$$W_c = \frac{WP}{150}, \text{ instead of } \frac{WP}{90}$$

This formula, changed as above, would, the writer believes, apply to the capacity of the starting-air compressor for starting Diesel-electric units, bearing in mind that starting the main engines of a direct drive necessitates the starting of an engine with the propeller load; whereas the starting of an electrically driven unit is accomplished at no load, it being necessary to overcome only the engine friction and generator and exciter windage.

The table given for the power required for various auxiliaries checks very closely with our experience on some points, particularly the power required for steering gear. It is assumed that the authors of this paper, although they have not so stated, have in all instances considered the power requirements of a hydroelectric steering gear. We had occasion to check this by having an ammeter directly connected to the steering-engine circuit during the first voyage of our largest vessel, and therefore have the actual operating figures. This is pointed out for the purpose of showing that this is not true of the so-called electric steering, where an electric motor is directly connected to a screw or like gear. Claims are made that the direct-connected steering gear in any one day does not consume the total kilowatts that a hydroelectric steering gear does. Having no accurate figures on the total power requirements of the electric steering gear, the writer has no means of refuting this statement that the actual power required for each movement of the rudder is very greatly in excess of the power required by the hydroelectric steering gear, and it is therefore seen that this power demand must be taken care of in available auxiliary generator capacity for this momentary load. This of course increases the cost of the Diesel-electric auxiliary generating unit, and the decreased cost of the direct-connected steering gear is thereby offset by the greater capitalization in the stand-by capacity required, not to mention the many mechanical advantages and the elimination of the electric starting contactors in the hydroelectric steering gear.

The foregoing comment is not intended in any way to criticize the valuable paper herein discussed, but to augment, if possible, the work of the authors by giving actual data which up to this time have not been available on the auxiliary requirements of electrically-propelled vessels.

SPENCER MILLER.[1] In Par. 8 the author says that when working cargo the electric winches do the same work as steam winches with a saving of approximately 90 per cent of the fuel. With Diesel-generated electric current this, in the main, may be correct. The author says the ship *Ashbee* burned in port, as a steamship, 7.6 tons of fuel, and after conversion to a motorship 1.43 tons, or one-fifth of its former fuel consumption, and this was accomplished with its original 8¼ by 8 steam winches. These particular winches were originally designed for the wooden-ship program during the war; they had reverse valves, and were designed not for fuel economy, but for simplicity, flexibility, and versatility, and to be handled by any class of labor, however stupid or untrained. The control was through a single lever, which was raised to raise the load, and lowered to lower the load. These steam winches are superior in speed to any electric cargo winch. Six thousand of these winches were built.

In another comparison by the author, two converted motorships, *Bidwell* and *Miller County*, retained their steam auxiliaries, and the fuel per day in port was three tons, whereas in the case of the electric auxiliaries on the *Challenger* the fuel consumption in port amounted to 1.33 tons. The cost of fuel is a small item in the cost of unloading and loading. The cost of handling miscellaneous cargo in the port of New York averages at least one dollar per ton. The fuel cost of the steam winch may be about three cents per ton, while that of the electric winch may be only one cent. From the standpoint of fuel cost, the electric winch will not have a very great influence on the cost of handling cargo. The fourteen cargo ships of the U. S. Shipping Board which are being converted to Diesel-propelled ships will carry highly developed electric winches which under test give the following results:

	Efficiency, per cent
Mechanical parts only	92 to 94
Hoisting:	
1,500-lb. loads	80
4,000-lb. loads	85
5,500-lb. loads	86
10,000-lb. loads	85

	Hoisting speeds ft. per min.
1,500-lb. loads	275
6,000-lb. loads	158
10,000-lb. loads	125

Comparing these speeds with the 8¼ by 8 winch referred to we find that 1500-lb. loads are hoisted 600 ft. per min., 6000-lb. loads about 250 ft. per min., and 10,000-lb. loads 100 ft. per min.

[1] Consulting Engineer, Lidgerwood Mfg. Co., New York, N. Y. Life Mem. A.S.M.E.

HARTE COOKE.[1] It has been the experience of the writer that electrically driven auxiliaries are not only very much more efficient than steam-driven auxiliaries, but that they also give less trouble and require less maintenance.

In regard to air-compressor capacity, the fundamentals of this should be seriously considered. The capacity of the compressor is reduced almost directly as the reduction in speed, while at slower speeds the volumetric efficiency would increase somewhat. This increase would by no means make up for the reduction in capacity, due to the lower number of revolutions.

The consumption of air, however, increases as the speed is reduced. While the fuel valves open a fewer number of times, they remain open longer, so that the total time they are open is about constant. Also, at the slower speeds, there is less fuel being used, which gives a chance for still more air to pass through the fuel valves. The practical effect of this is shown on the curves in Fig. 4. With a reduced delivery of air and a constant time of opening for the fuel valves, the injection pressure will naturally decrease.

From the above it can easily be seen that the larger the capacity of the compressor, the higher the injection pressure that can be maintained at slow speeds, and with certain fuels this compressor capacity would undoubtedly limit the minimum speed at which the engine could be operated. With a marine compressor which is provided with a proper unloading device, the normal demands for air always can be met with a good mechanical efficiency, and therefore any excessive reduction in the capacity of the compressor to reduce the weight and cost tends to limit the ability of the engine to perform at the slower speeds.

MARTIN L. KATZENSTEIN.[2] This paper presents the many phases of selection and design of motorship machinery, and is apparently the result of practical experience by both authors.

In Par. 14 reference is made to moderate-speed versus high-speed auxiliary generating sets. The authors do not say definitely what constitutes low speed. It would seem that simplicity of construction should be the deciding factor, for with it, with an engine properly designed, low maintenance costs are to be expected. For the cargo vessels that are discussed in this paper, the size of auxiliary units never exceeds 100 b.hp., and speeds below 350 r.p.m. would result in designs excessively heavy, bulky, and correspondingly high in cost.

Where air-injection engines for auxiliary service are used, reference is made elsewhere in the paper to oversize air compressors for supplying maneuvering air for the propelling engine.

[1] McIntosh & Seymour Corporation, Auburn, N. Y. Mem. A.S.M.E.
[2] Manager, Marine Department, Worthington Pump & Machinery Corporation, New York, N. Y. Mem. A.S.M.E.

Such an arrangement involves the running continuously of large low-pressure pistons, whether they are required to compress air or to idle, and thus there is a continuous waste of power. A simple (though not compact), economical, and reliable combination for auxiliary engines, generators, and maneuvering air compressors consists of (*a*) a solid-injection engine running at a speed of from 300 to 375 r.p.m., in powers ranging from 60 to 120 hp., direct connected to (*b*) a generator, and driving through a magnetic clutch fitted to the extended generator shaft (*c*) a maneuvering air compressor. The magnetic clutch, controlled from the propelling-engine operating platform, permits driving the compressor or leaving it at rest, as requirements for air may demand.

This arrangement, then, eliminates a separate prime mover for the maneuvering compressor, which when air is not required is at rest and not wasting power.

Magnetic clutches are entirely reliable in service, and such combinations as are here described can be seen on a number of American-built or converted motorships, such as the *Steel Motor* and *Steel Vendor*, operated by the U. S. Steel Products Corporation, the *Challenger*, etc.

If the auxiliary engines are of a port-scavenging, two-cycle design, provision can be made for burning the same fuel as is used in the propelling engines. Such designs are probably the simplest, so far as running gear is concerned, that can be found anywhere, and a minimum of attention by the operator is required.

The authors, in Part II, refer to the fixed charges against the excess cost of Diesel machinery for ships. This is true and has been one of the real causes for retarding the development of the motorship in this country. It is stated that the disadvantage of high cost diminishes in proportion to the time that the machinery is kept at work. This has two applications, i.e., in long voyages (with quick turn-around) and in utilizing the running gear of the propelling engines as great a part of the time as possible. Our foreign competitors are better placed than we for long voyages, and this would account for the greater progress in motorship development abroad. To overcome this we have turned to the development of the double-acting, two-cycle oil engine, and on this subject a paper[1] was presented before the Society in December, 1924, by Dr. C. E. Lucke, in which the advantages of that type were set forth in considerable detail.

So far as producing power is concerned, the double-acting two-cycle engine functions just as does a reciprocating steam engine, and by utilizing the running gear for every stroke, power is produced at a lower cost than is possible with single-acting engines. The U. S. Shipping Board is now carrying out the conversion to motorships of fourteen steam-propelled vessels. It is known generally that of bids received on all types of engines from many

[1] Trans. A.S.M.E., vol. 46 (1924), p. 1005.

points in the United States, the lowest in weight, dimensions, and cost were for the double-acting, two-cycle types.

The first of these engines will soon be ready for installation and trial, and it is believed that with a successful demonstration the use of such engines will make possible the economical conversion of existing steam vessels to motorships or the building of new vessels to take this most modern form of propelling machinery.

JOSEPH HECKING.[1] The authors stress the necessity of the larger number of auxiliaries required for Diesel ships than for steamships. Whatever criticism in this direction is real and justified is due to the extreme caution of engine builders and classification societies in not taking chances on the complete success of the marine Diesel engine in its infancy. The effect of their caution is seen in the rapidly increasing number of Diesel-engined ships.

In a general comparison of auxiliaries for steam and Diesel plants we must not forget to include in the steam plant the boilers and their accessories, the condensers, main and auxiliary, and accessories, and the comparatively large and expensive piping systems. If it were true that the Diesel plant has a larger number of auxiliaries than a steam plant, it would not be possible to operate the former with a considerably smaller engineers' crew than required for a steam plant. It would be interesting to learn from an authoritative source the comparative cost of the steam auxiliaries, including boilers, with the auxiliaries of Diesel ships. The writer is convinced that the excess in cost of the Diesel engine installation over the steam plant is entirely due to the greater cost of the main Diesel engine. When we speak of Diesel engines we have in mind the fully developed four- and two-cycle single-acting air-injection engines. While the solid-injection engine has been fully developed in small and medium sizes, the upper limit in their development has not as yet been reached; nor do we know today the limits of the double-acting two-cycle engine. Both developments, if carried out successfully, should have in their wake a reduction in the first cost of the Diesel plant.

In Par. 13 the authors find that usually not less than three Diesel generator sets are fitted. This should probably be "not more than" three sets. In vessels of short coastwise routes, where a fixed sailing schedule with little time in port must be maintained, it may be desirable to carry a spare generator set, but for long-distance routes it looks like a waste of money and tonnage, and the total kilowatts need not be greater than the maximum port load, especially where three sets are fitted. It is impossible to lay down hard-and-fast rules; every ship presents its own problem. It required not more than a half a dozen installations to demonstrate the folly of placing steam auxiliaries on Diesel ships, and even the suggestion would appear like a backward step.

[1] Engineer-Surveyor, American Bureau of Shipping, New York, N. Y.

In Par. 63 the authors speak of the handicaps by classification societies. This remark is partly justified, but the rules of the American Bureau of Shipping were developed with the advice and coöperation of all important engine builders in this country, and the various restrictions have prevented enthusiastic engineers from spoiling the success of the marine Diesel engine. The writer remembers well the emphatic remarks of an engine builder to the effect that no auxiliary compressors would be required with his engine due to the perfect design of his main compressor. It later developed that the vessel fitted with his engine had principally compressor trouble and was brought safely into port by the auxiliary compressor. Furthermore, salt-water jacket cooling, fresh-water or oil piston cooling, and pressure lubrication are not requirements of the classification society, but entirely builders' choices. Where such features are considered necessary it is obvious that the necessary pumps should be fitted to perform the work.

The paragraphs, tables, and curves pertaining to air compressors deserve particular attention. It is impossible to form definite conclusions from a tabulation of compressor dimensions as actually fitted aboard the ship for the reason that engine builders cannot economically design a new compressor for a considerable range of powers. The values given by Güldner seem to apply very satisfactorily; that is, the ratio of cylinder volumes to low-pressure compressor volume equals ten for two-cycle engines and twenty for four-cycle engines. In an article in the Bulletin of the American Bureau of Shipping the writer emphasized the desirability of fuel-valve lift control on marine engines, and it is gratifying to note that the engineers of the U. S. Shipping Board have incorporated a corresponding requirement in their specifications for Shipping Board engines. The effect on the size of the attached main compressor is readily understood, when it is considered that the quantity of air flowing through a fixed orifice is dependent on the period of valve opening. At one-third engine speed the delivery capacity of the attached compressor is about one-third of normal, whereas the engine consumption is the same as for full speed. With fuel-valve lift control the supply and consumption of injection air can be manipulated so as to be nearly constant.

With regard to the reference to the American Bureau rules for compressor capacities the writer has to state that we have endeavored to make the requirements adequate and clear, but we are aware that conditions may arise where the definitely stated requirements do not apply, and such cases will be considered on their merits. The underlying principle of all classification rules is the safety of ship and cargo. In the case of a low-speed cargo vessel the capacity of the auxiliary compressor is apportioned so as to permit of about three-quarters normal speed, which is considered the minimum seaworthy speed in heavy weather.

HAROLD ANDERSON.[1] It is rather interesting to see the favorable viewpoint the author takes toward the waste-heat boiler. One reason for the usual prejudice against this type of boiler is probably the prevailing opinion that inasmuch as the exhaust temperature from a Diesel engine is practically the same as that of the flue gases from a boiler, there is no justification for the installation of a waste-heat boiler. The fact that in land practice economizers are used in connection with boilers, bringing the flue gases down to from 200 to 300 deg. fahr., is apparently lost sight of when the waste-heat boiler is condemned.

One application of waste-heat boilers is described which makes use of steam-driven auxiliaries. It seems, however, that the better application would be to use the steam so generated in a steam-driven generating set, which in turn would supply the electrically driven auxiliaries. This steam-driven generating set could either be condensing or exhaust into the ship's heating system. This installation would make it possible to have the auxiliary machinery all electric, thus simplifying the engine-room arrangement, and also effect a saving of fuel of possibly 5 per cent. Diesel-driven auxiliary generators, of course, would also be installed to act as spares and for port use.

In Par. 113 it is suggested that the ship may be heated electrically. While this is very convenient for remote places and attractive in passenger service, it should be kept in mind that electric heating for this purpose is expensive. Assuming that the auxiliary-electric-generator engine requires 0.5 lb. of oil per b.hp-hr. or, say, 0.75 lb. per kw-hr., it is obvious that the use of one pound of fuel would give only $3412 \div 0.75 = 4550$ B.t.u. at the point of use. If we assume an oil having a heating value of 19,500 B.t.u., 4550 divided by 19,500, or 23.3 per cent, would be the percentage of heat transmitted which is equivalent to the boiler efficiency. As an oil-fired boiler may easily be kept at 70 per cent efficiency, thus trebling the heat delivered if transmission losses are excluded, steam heating is apparently the most economical solution. The use of steam, as outlined above, would also give the incidental advantage of allowing its use for preliminary warming up of the engine, as mentioned in Par. 96.

The final criterion, however, is the ultimate cost of operation, and the initial cost may perhaps more than offset any theoretical gain in efficiency.

WILLIAM H. THOMPSON.[2] The interests of the writer's organization are centered in the exhaust-gas generators or boilers, and his remarks will be confined to this equipment. The first item for

[1] General Engineer, Westinghouse Electric & Mfg. Co., South Philadelphia Works, Philadelphia, Pa. Assoc-Mem. A.S.M.E.

[2] Chief Engineer, Davis Engineering Corpn., New York, N. Y.

consideration is whether or not the boiler should be installed. If there is any demand for heat, then a boiler should be considered. If there is no demand for heat, but the engine is of large power, then consideration should be given to the additional power that could be obtained by steam generated by the exhaust gases. If heat only is required, it must be borne in mind that every 600 B.t.u. saved from the exhaust gases would mean an expenditure of from 900 to 1000 B.t.u. if oil were to be burned under a boiler. That ratio must be the basis for arriving at the savings. The same ratio applies to cases where the steam is generated for developing additional power.

The method of determining the approximate saving in B.t.u. has been outlined by the author. The final figures should be determined in consultation with the manufacturer of the boiler, the design of which is important. The maximum of efficiency is obtained through the proper ratio or area of exhaust-gas passages to length of travel. Our experience has shown that the ordinary boiler does not meet these requirements.

In our opinion we consider the fire-tube type the design which can be figured most accurately. It also permits of more thorough cleaning of soot deposits. These boilers must be cleaned from time to time, as there is bound to be a deposit of soot on the heating surface just the same as in any boiler fired directly with fuel.

It has often been asked if the exhaust gases from Diesel engines will corrode the tubing. We had the chance of observing one of our generators that had been in service for nearly two years, and after it had been cleaned there did not appear to be any effect of gases on the tubing.

These boilers can be so proportioned that they will add no appreciable resistance to the discharge of exhaust gases. In fact, they tend to decrease the back pressure on the engine, because the gases decrease in volume due to lowering of the temperature through the boiler. We have also found that the exhaust gases from two engines can be discharged into one generator, or boiler, at the same time without having any appreciable effect on the working of either engine.

It is not practical, as far as our investigations have proved, to have the exhaust gases and an oil burner working in the same boiler at the same time. Furthermore this would not be an efficient way to operate. By simply bearing in mind that the exhaust gases are at a temperature considerably lower than the flame of the burner, it will be realized that if the two gases were mixed they would have an average temperature much lower than the flame temperature, and therefore necessitate considerably more heating surface to absorb the heat.

If it is necessary to develop more steam than can be obtained from the exhaust gases, the more practical and efficient method is to have two separate boilers, one for the exhaust gases and one

for burning fuel. If steam is required practically all the time, and a sufficient amount can be obtained from the exhaust gases when the engine is in operation, then the boiler should be equipped with a suitable firebox so that the oil can be burned under it when the engines are not in service.

A. W. ROBINSON.[1] One of the most promising fields for Diesel-electric power is that of dredging machinery. In this class of machinery almost the whole power may be said to be consumed by auxiliaries. We have power for hoisting, swinging, working mooring lines, and various other things. Electrical distribution of power is admirably adapted to that purpose. In the writer's search for a suitable oil-electric engine to do this kind of work, he has been confronted with a choice between a large engine, slow moving, adapted for screw propulsion, and a very small type of high-speed engine of the racing machine or aircraft type, neither of which is suited for the purpose in hand.

A large dredging machine which is now being built at the works of the New York Shipbuilding Company will handle a 10-yd. dipper to a depth of 52 ft. and will have a direct pull of 300,000 lb. In the search for a suitable oil engine to drive this machine, we were compelled to fall back upon a slow-moving type of engine, whereas we much would have preferred to have employed a smaller, lighter, faster engine, suitable for electric generation of power, if such a one could have been found. However, we were unwilling to tie up to an engine of the aircraft type or racing-machine type, which is very small and very light and requires an expert to run it, and we did not feel that such an engine could be run by the ordinary marine engineer.

Therefore, the writer commends to the Diesel-engine manufacturers the desirability of producing something between the ponderous, slow-moving marine type, and the fast-running aircraft type, of a kind that will be suitable for electric generation of power and easy for the average marine engineer to operate. An engine has recently been brought out in Europe of the high-speed type, which is a beautiful piece of workmanship. It runs at about 600 r.p.m. and weighs about 10 lb. per hp. What we want is an engine of rugged type and simple construction that any marine engineer can run, and which will be suitable for the direct generation of electric power.

JOHN W. MORTON.[2] The author takes this opportunity of thanking the large number of contributors to the discussion for their exceedingly valuable comments. The discussion covers such a wide

[1] Consulting Engineer, Upper Melbourne, Province of Quebec, Canada. Mem. A.S.M.E.

[2] Author's Closure.

field that it has been considered advisable to reply to each contributor individually.

It is noted with pleasure that the problem of auxiliaries for motor vessels has appealed to engineers in other branches than that in which the author is engaged.

Mr. Mulheron's remarks are of great interest and the author is in general agreement with him.

The voltage 110 and 220 is perhaps antiquated, although most European ships operate with this voltage, but the author is aware that a higher voltage is used in the United States although the difference is small measured at the point of use.

The question of air-compressor capacity for the Diesel-electric form of propulsion would naturally come under the category of "stationary plants," the starting of which can be accomplished by a "kick" from one or two cylinders, using high-pressure air from small reservoirs that are charged during the operation of the engines in the case of a directly attached air compressor, or from a small auxiliary air compressor.

In regard to compressors for scavenging purposes with two-cycle engines, Dr. Sanford A. Moss covered this very thoroughly in his paper.[1] As this kind of compressor (reciprocating or turbo type) only works with low pressure (from 1 to 3 lb. over and above atmospheric), its volumetric displacement is generally around 1.4 times the volume of the working cylinder.

Receivers are generally employed, and the compressor is frequently made double-acting to insure a steady, uniform-pressure flow.

Referring to the discussion by Mr. Shepard, a conclusive verdict cannot be reached, considering the vast number of motor vessels, with almost as many varieties of engine installations, each based on the experience of some chief engineer; for it can hardly be expected that any two engineers will plan exactly the same layout of an engine room. Conservatism is a good policy, where conditions demand it, and the author might cite the case of a great European oil-engine and shipbuilding concern that only recently considered the matter of reverting to the use of steam winches on a motorship.

In regard to performance of the winches under discussion, the author knows of several recent installations where all winches on lowering cargo returned this potential energy to "Storage," and this really constituted an efficient auxiliary plant. This of course is only possible with electric winches.

Mr. Miller's remarks are of great value, and the author recognizes his long and vast experience in this particular field. The author wishes to state that the N. Y. Shipbuilding Corporation had a specially designed piston-type reverse valve fitted to the

[1] Centrifugal Compressors for Diesel Engines, *Mechanical Engineering*, vol. 47, no. 11a, Mid-November, 1925, p. 1075.

winches mentioned on the *Ashbee* and *Jacksonville* (sister ships) with very good results. These winches are subject to very rough handling by the natives of the west coast of South America.

Replying to Mr. Goldsmith, it must be borne in mind that when the installation is of the Diesel-electric type, we really have a "floating" stationary plant with one control station with a switchboard from which all energy is distributed, whether it be for propulsion or for auxiliary purposes. Nevertheless conditions will naturally differ to a certain extent, according to whether the engines are of the air-injection or airless-injection type, and due allowance in auxiliaries should be provided.

As to the compressor capacity, the author's reply to Mr. Mulheron holds good. Full-Diesel engines (with high compression) of the stationary type are generally started by air of the same pressure as the injection air used, and consequently require only small-capacity air reservoirs. Starting is effected on multiple-cylinder engines (4-6-8) on only a couple of cylinders, the engine being barred over to the right starting position, and the starting consists merely of a good "kick-over." Some engines are equipped with an air-starting distributing system, and as engines also vary in their compression ratio, due allowance for all these factors must be made when calculating the capacity of the auxiliary air-compressor capacity, as well as the air reservoirs. Still, Mr. Goldsmith's remarks are very valuable, as really no definite information has been available on Diesel-electric vessels.

Mr. Cooke's remarks, especially on air-compressor capacity, are well founded, and confirm the author's conclusions as revealed both in the text and by the diagrams.

In reply to Mr. Katzenstein's remarks concerning moderate speed, the author would suggest one of about 300 r.p.m.; but he is aware of the many factors entering into the pros and cons of the still higher-r.p.m., short-stroke, and the lower-r.p.m., long-stroke types of engines. Each has its advantages and disadvantages as to weight, space, cost, etc. Simplicity is a matter of design, a trunk-piston engine naturally being simpler than one having a crosshead, and a "port" scavenging two-cycle engine simpler than a "valve-in-the-head" type. But as the present market swarms with a multitude of designs, it is in the last analysis the owner who is the judge, and he will naturally install an engine which to his knowledge (or to that of his chief engineer) is the one best suited for his purpose. A multitude of variations in auxiliary layout can be found in present-day practice, including the one to which Mr. Katzenstein refers; even chain drive from the main engine has been resorted to in some cases.

In reference to Mr. Hecking's discussion, the author knows quite well that a large factor of safety is required for a marine plant, but this can also be overdone in some cases. Auxiliary equipment on marine plants is generally installed in duplicate; nevertheless

it has been the experience of the author that most of the trouble on board ship has been due to the auxiliary equipment and not to the main propelling plant, although the latter works day in and day out, 24 hours a day, whereas the auxiliaries are generally working only half this time, or intermittently. The author might be an idealist but for the fact that when reviewing the hundred or more years of experience engineers have had with the simple steam plant, he finds that there has been only a very slight gain in efficiency and this only at the expense of ever so many varied auxiliaries introduced in the modern power house, thereby increasing the initial cost. We are still using water to raise steam, just as James Watt did, only in a more elaborate way. Due to the boilers, it is a matter of consequence that a larger crew has to be maintained on a steam vessel than on a motorship, even if the boilers are equipped with all up-to-date labor-saving devices, or are of the oil-burning type.

The type of vessel to be built or converted, of course, has a large influence on the selection of the main propelling plant and its auxiliaries. In all cases every factor should be considered carefully beforehand and the selection founded upon sound engineering experience.

The author has already pointed out in this paper the benefit to be derived from automatic or semi-automatic fuel-valve-lift variation as a great saver of costly compressed air. This becomes ever so much more a necessity when operating engines of the slow-speed, long-stroke type.

Replying to Mr. Anderson, on the motorship *Gulfcrest*, built by the New York Shipbuilding Corporation for the Gulf Refining Company, the exhaust gases from the two main engines can be bypassed into the two Scotch boilers installed on board. On test runs the steam raised was sufficient to run all engine-room auxiliaries, including the steam-driven generating set and the heating system (about 3000 lb. of steam raised per hour, which checks up with the author's figures).

The author agrees with Mr. Thompson's remarks as to whether or not a boiler should be installed. Each case demands thorough investigation.

Another thing to be considered in an exhaust-gas heating-boiler arrangement besides the length and area of exhaust passages, is the attendant noise produced. The wave problem here is analogous to that in the pipes of an organ, and should be well considered in laying out the exhaust passages, as also should be proper expansion joints, their lengths and construction.

Mr. Robinson, if he continues his search for proper machinery for dredges, will find that there are a few manufacturers in the United States who are building oil engines that just fit in between the extreme types he mentions, both of the air-injection as well as the airless-injection type, and of moderate speed.

No. 1985

THE GAS ENGINES OF THE MARYLAND PLANT OF THE BETHLEHEM STEEL COMPANY

By A. A. Raymond,[1] Sparrows Point, Md.

Non-Member

This paper presents details of an installation at the Sparrows Point plant of the Bethlehem Steel Company of eighteen large engines using blast-furnace gas. It includes tables showing results of tests, operating records, and the amount of total heat available from exhaust gases of blowing and generator-unit engines.

IN THIS plant, located at Sparrows Point, Md., there are eighteen gas engines operating on blast-furnace gas, twelve blowing engines and six engines driving electric generators. These eighteen units are housed in three buildings, connected to permit easy transfer of material. Each building is equipped with an overhead electric crane, two of them being of fifty tons capacity and one of forty tons.

2 The blowing engines are two-cylinder four-cycle double-acting. The first five, installed in 1914, have gas cylinders 45 in. in diameter and air cylinders 80 in. in diameter, and the seven added since have 47-in. gas cylinders and 84-in. air cylinders; the stroke of all the engines is 60 in.

3 The generating units each consist of a four-cylinder twin-tandem engine having cylinders 47 in. in diameter and 60 in. stroke directly connected to a 25-cycle 6600-volt generator running at 88.3 r.p.m. and rated at 4000 kw. at 80 per cent load factor.

4 The fuel gas after passing through the wet washers at the furnaces is carried to the Theisen washers, of which there are eight distributed in the three buildings, each having a capacity of 850,000 cu. ft. per hr. Each washer is driven by a 150-hp. motor running at 355 r.p.m., and all discharge into a common clean-gas main from which there is an engine supply line to each building. Connected to the clean-gas main is a gas holder having a capacity of 202,000 cu. ft., the flow of gas to and from the

[1] Chief Engineer, Gas Engine Department. Deceased, January 26, 1926.

Contributed by the Oil and Gas Power Division for the Annual Meeting, New York, November 30 to December 4, 1925, of The American Society of Mechanical Engineers.

holder being regulated by an automatic valve operated by a gasometer. This arrangement permits the successful operation of blowing engines in a one-furnace plant, the holder supplying gas to the blowing engines for the short time that the blast is off the furnace at casting time. This plant operated during 1921 with but one furnace in blast and produced 58,000,000 kw-hr. with the gas-driven generators, with a maximum hourly output of 1400 kw.

5 The total water required for the engines amounts to 316,600 gal. per hr., of which approximately half is salt water used for gas cleaning, the blowing engines using 4800 gal. of cooling water per engine-hour and the generator units 15,000 gal. per engine-hour. After passing through the engines the water is cooled by spraying, the average temperature of the water to the engines being 85 deg. fahr. and the leaving temperature 145 deg. fahr.

6 For starting the engines, air at 200 lb. pressure supplied by motor-driven compressors is used. It is stored in fifteen tanks located in the basement, and supplied to all engines from a common line.

Details of Engine Construction

7 *Cylinders.* The engine cylinders are made of cast steel in either the one- or two-piece type. Steel is used in preference to cast iron for the following reasons:

a Owing to its superior strength, the walls can be made thinner than when cast iron is used, which permits a more rapid transfer of heat and more uniform cooling of the metal throughout.

b The limit of thickness is not strength, but that necessary to make a good casting. This is especially of advantage in large dimensions where the thickness, if cast iron were used, would be prohibitive.

c Steel is a better metal than cast iron to withstand, without fatigue, shocks due to explosion and also to withstand temperature stresses.

d Steel can be put in tension with safety.

e Owing to permissibly thinner steel cylinders, the wall temperature can be maintained at a satisfactory point with jacket water at a temperature much hotter than that allowable with cast-iron cylinders. This increases the thermal efficiencies, especially in large cylinders.

f The use of steel permits designs of valve ports and other details which decrease the overall length. Bethlehem engines as compared to other makes of gas engines of the same horsepower are much shorter.

g Steel permits a lighter engine and at the same time a much stronger one than cast iron.

8 *Pistons.* The pistons are of cast steel, cast in one piece without ribs. As far as possible the shape of the piston is spherical. The piston is thus safe against temperature stresses and rigid

against explosive pressures. Having comparatively thin walls as compared to cast-iron pistons, there is better cooling. The weight is less than that of a cast-iron piston and there is consequently less piston-rod deflection. This also decreases the weight of the reciprocating parts, thus allowing speeds higher than would otherwise be safe. The packing rings are very deep, which prevents wear on the sides of the grooves. The piston is pressed on the rod and held in place by a nut which is doweled and staked.

9 The piston rod is bored for water cooling from both ends to avoid using any plugs. The water being introduced into and taken from the ends of the rod, it is unnecessary to drill transverse holes in the ends, thus avoiding any weakening of the rod at these points. The holes for the water inlet and outlet from the piston are drilled where the rod is largest in diameter and at an angle, so that a section through the rod where the holes are drilled is no weaker than any other section. Both rods are alike and are thus interchangeable by turning them end for end.

10 The piston-rod packing is designed with rings of the segmental type, floating on the piston rod. They are kept in place by springs having physical properties adapted to resist the effect of heat. There is no complication due to water cooling. The lubrication is excellent as the rod is oiled by droppers from the top. The packing is easily cleaned and inspected. All packing rings are made of cast iron.

11 *Lubrication.* A continuous-pressure oil system is used wherever possible. Each engine has its own oil pump driven from the end of the layshaft. Engine oil is supplied under pressure to the main bearings, crankpins, and crosshead pins, and to the main and intermediate crosshead sides. From these the oil flows to the filters. Each engine having its own filter and pump, it is independent of any other engine in the power house.

12 Cylinder oil is supplied to the cylinders, piston packing, and exhaust-valve stems by sight-feed oil pumps which are also driven from the layshaft. There are two inlets at each end of each cylinder, thus insuring a good and even distribution of oil on the piston rings. The time of injection of the oil is such as to thoroughly lubricate the cylinder before each explosion stroke.

13 *Governing.* The governor, of the standard spring-loaded type, is driven by gears. It does not act directly on the mixing valves but controls an oil-operated piston by means of a relay valve in hydraulic balance. The work required of the governor is constant regardless of the load of the engine or the physical condition of the mixing valves. This arrangement secures more rapid control under all operating conditions, which is extremely important for electric generators operating in parallel.

14 The mixing valve gives a mixture of absolutely constant quality under all loads. The amount of mixture is varied with the

load, and at no time is the amount of gas so small as not to form a combustible mixture. By having the ports cut to 45 deg. an effect similar to a cut-off valve is obtained, yet at the same time

TABLE 1 DATA OF TESTS AND OPERATING RECORDS OF GAS ENGINES AT THE MARYLAND PLANT OF THE BETHLEHEM STEEL CO.

Blowing Engines	
Average free air per minute required per furnace, cu. ft.	42,000
Average blast pressure, lb. gage	20
Engines per furnace	2
Revolutions per minute per engine, 80-in. air tub	69
Revolutions per minute per engine, 84-in. air tub	62
B.t.u. per cu. ft. of dry fuel gas	105
Cu. ft. gas consumed per engine-hour	145,000
Total heat consumed per engine-hour, B.t.u.	15,244,000
Thermal efficiency per i.hp. per hour in gas cylinders, per cent	28
Heat Balance	
Heat carried off by cooling water, per cent	27.4
Heat in exhaust gas, per cent	36.7
Heat unaccounted for, per cent	7.9
Heat converted into work in cylinders, per cent	28.0
Total hourly gas consumption of blowing engines, cu. ft.	1,745,000
Generator-Unit Engines	
Kilowatts per unit per hour	3400
B.t.u. per cu. ft. of dry gas	105
Cu. ft. of gas consumed per engine-hour	481,000
Cu. ft. of gas consumed per kw-hr.	143
Total heat to engine per hour, B.t.u.	50,524,430
Heat Balance	
Heat carried off by cooling water, per cent	28.0
Heat carried off by exhaust gas, per cent	38.0
Heat lost and unaccounted for, per cent	5.4
Heat converted into work in the cylinders, per cent	28.6
Total hourly consumption of generator-unit engines, cu. ft.	3,000,000
Total hourly gas consumption of blowing and generator-unit engines, cu. ft	4,745,000

the valve action is positive. The same design of valves is used for all gases.

15 The mixture is ignited by three spark plugs on each cylinder, of the make-and-break type, mechanically driven. Current is supplied at 110 volts d.c. Induction coils are used to increase the density of the spark.

Results of Tests and Operating Records

16 Tables 1 and 2 are the results of tests and operating records. Of the total heat available from the exhaust gases the amount

TABLE 2 TOTAL HEAT AVAILABLE FROM EXHAUST GASES OF BLOWING AND GENERATOR-UNIT ENGINES

Blowing Engines	
Total heat to engines per hour, B.t.u.	15,244,000
Total heat in wet exhaust gas per hour, B.t.u.	6,000,000
Temperature of exhaust gases, deg. fahr.	900
Heat convertible in waste-heat boilers, B.t.u. per hr.	3,000,000
Boiler hp. available per hour	88.2
Generator-Unit Engines	
Total heat to engines per hour, B.t.u.	50,524,000
Total heat in wet exhaust gas per hour, B.t.u.	19,000,000
Temperature of exhaust gases, deg. fahr.	1,000
Heat convertible in waste-heat boilers, B.t.u. per hr.	9,500,000
Boiler hp. available per hour	279.4
Exhaust heat from generator-unit engines utilized, per cent	6.5

now utilized is very small. Besides the hot water furnished to the boilers from the circulation system, sufficient low-pressure steam has been generated during the winter months to supply heating coils in the oil filters and cylinder-oil pumps and furnish some heat in the buildings. This steam is generated by the use of heating coils placed in the exhaust headers, the coils being supplied with water leaving the engines at about 145 deg. fahr.

Total steam generated, boiler hp-hr. per month	57,000
Total steam generated, boiler hp-hr. per engine-hour	19

Table 2 shows the total heat available from exhaust gases.

No. 1986

RECENT DEVELOPMENTS AT COLFAX STATION, DUQUESNE LIGHT CO.

By Charles W. E. Clarke,[1] New York, N. Y.

Member of the Society

This is the fourth of a series of papers dealing with the Colfax Station of the Duquesne Light Co., Pittsburgh, Pa., which the author has presented to the Society. In the present paper are given the test results of the 3-A element of the station which showed net heat rates of 12,750 B.t.u. per kw-hr. at 30,333 kw., 13,021 B.t.u. per kw-hr. at 22,400 kw., and 14,200 B.t.u. per kw-hr. at 15,050 kw. A brief history of the physical plant involved in the original station, and in the unit extensions Nos. 1, 2, and 3, is given, with data on boiler and turbine outages and condenser cleaning. The author shows that the total cost of the station has been to date $115.80 per kw. for the 190,000 kw. of installed capacity, and predicts a total cost, when the fourth unit is installed, of $107.50 per kw. for 270,000 kw.

Physical Development

THE original installation at Colfax Station was put in operation on December 18, 1920, and consisted of one 60,000-kw. 3-element turbo-generator unit, equipped with 100,000 sq. ft. of condensing surface in four shells, with LeBlanc air pumps. The boiler equipment consisted of seven cross-drum boilers, each 51 tubes wide and 18 tubes high, with superimposed superheaters, fired by 17-retort, 22-tuyere underfeed stokers with double-roll clinker grinders. The operating pressure was 275 lb. per sq. in. and the total temperature 600 deg. fahr.

2 The heat balance or auxiliary power system for the first unit consisted of a house turbine of 2000-kw. capacity, operating on the house-service bus, supplemented by a 750-kw. synchronous induction motor-generator set which is connected through an auxiliary house bus to the main unit. The exhaust from the house-turbine unit was condensed in atmospheric jet condensers or feedwater heaters. A complete description of this system was given in a paper

[1] Dwight P. Robinson & Co.

Contributed by the Power Division and presented at the Annual Meeting, New York, November 30 to December 4, 1925, of The American Society of Mechanical Engineers.

presented to the Society at the annual meeting of 1921, Heat Balance of Colfax Station.[1]

3 The second 60,000-kw. unit was practically a duplicate of the first unit, with the exception that steam-jet air pumps were provided in connection with the condensing equipment and six boilers, 20-tubes high, were installed instead of the seven 18-tube-high boilers of the first installation. Superheaters were of the interdeck type. This extension was placed in operation October 21, 1922.

4 In connection with the second-unit extension, extensive outdoor coal-handling and storage equipment was provided, consisting of a traveling, rope-operated bridge, 330-ft. span and 1000-ft. travel, with an average capacity of 267 tons per hour. Shortly after the second-unit extension was completed, it was thought desirable to install air preheaters on several of the boilers, and the first unit so equipped was put in operation on May 7, 1923. Since that time three stoker-fired boilers have been provided with air preheaters. The results of this development are covered in a paper presented at the annual meeting of the Society in December, 1923, Boiler Test Results with Preheated Air, Colfax Station, Duquesne Light Company.[2]

5 At the time of the third-unit extension in 1923, the questions of higher pressures and temperatures, pulverized fuel versus stoker firing, preheated air, multi-stage bleeding, and steam reheating at a middle point of the turbine were all discussed. It was decided to maintain the same operating pressure, increase the temperature to 650 deg. fahr., adopt pulverized-fuel firing in place of stokers and install 4-stage bleeding in connection with an auxiliary generator attached to the main-turbine shaft as a method of heat balance. It was also decided to install two turbine units in place of one larger unit.

6 The third-unit extension as now constructed consists of enlarging the present turbine and electrical bays sufficiently to house two 35,000-kw. multiple-exhaust turbines, together with the transformers and bus and electrical connections necessary for these two units, and increasing the boiler house to take five boilers.

7 One 22,914-sq. ft. stoker-fired boiler, similar to those furnished with the second unit, was installed to fill in a vacant space in this section of the boiler house, and five 27,680-sq. ft. pulverized-fuel-fired boilers were installed in the third-unit extension. These latter boilers are 47 tubes wide by 20 tubes high, with 24-ft. tubes, and are equipped with fourteen burners per boiler, set in two batteries of seven each. The drums are set five feet higher than those of the first and second units in order to provide increased combustion space. The superheaters are of the interdeck type. These boilers were made 47 tubes wide in order to provide room

[1] Trans. A.S.M.E., vol. 43 (1921), p. 487.
[2] Trans. A.S.M.E., vol. 45 (1923, p. 567.

to carry the preheater air ducts down the side of the boiler walls and still maintain the same column centers as exist on the original installations.

8 The pulverizing and fuel-feeding equipment is compactly arranged above the aisle between the two rows of boilers. Two

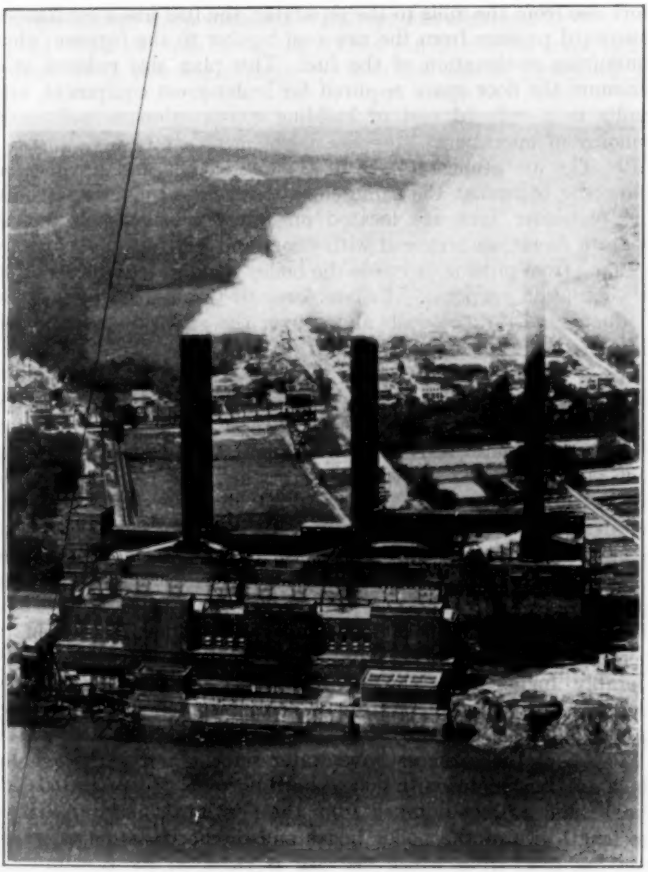

FIG. 1 THE COLFAX STATION AS IT IS TODAY WITH THE THIRD-UNIT EXTENSION

six-ton mills are installed with each boiler. These are located on the top floor of the boiler house directly under the raw-coal bunker and at an elevation 16.25 ft. above the boiler drums; the separators are above the mills and the mill and separator room is completely walled off to isolate it from the remainder of the boiler house.

9 Pulverized-coal bunkers of 35 tons capacity are installed in front of each boiler above the drum line, and these supply fuel directly to the fourteen burner feeders attached to the boiler. The bunkers are fed through a duplicate set of screw conveyors, cross-connected so that any bunker may be fed from any set of mills on either side of the house. It will be noted that except for the short rise from the mills to the separator, the fuel has a continuous downward passage from the raw-coal bunker to the furnace; thus minimizing re-elevation of the fuel. This plan also reduces to a minimum the floor space required for boiler-room equipment, and results in a reduced cost of building construction as well as in economy of operation.

10 The air preheaters are of the plate type and are located above the boilers at the same elevation as the pulverizing mills. The preheater fans are located on the same elevation but in separate rooms, so arranged with doors and shutters that air may be taken from outside or inside the boiler room as may be required for ventilation purposes. The fans force air through the preheaters, air ducts, and furnace walls and deliver the air to the combustion space of the furnace at about zero pressure.

11 Two of the boilers have furnaces constructed with hollow side and front walls through which the preheated air passes before it enters the furnace. The rear walls of these furnaces are protected by plain-tube water screens. These boilers have been in operation since October, 1924, and the furnaces are in good condition. It has been found, however, that with CO_2 higher than 13 per cent, the furnace walls will slag and run. The operation of these boilers was discussed by the author at the Milwaukee meeting of the Society in the spring of 1925. The other three boilers have furnaces constructed with fin-type water screens on the side, plain tubes on the rear walls, and hollow front walls. They have been in operation since July, 1925, and the indications are that a considerably higher CO_2 can be maintained without appreciable slagging. Fin-type coolers are now being installed in the side and rear walls of the first two boilers.

12 All of the furnaces have water screens across the ashpits which are constructed with hollow-lined bottoms and are air-cooled. Clinker and ash are dumped from the pits into sluiceways which run lengthwise of the boiler house and discharge into transverse receiving pits, from which the ash is removed by a clam-shell bucket and delivered directly to distributing cars or trucks.

13 The two turbo-generators which comprise the third-unit installation are placed crosswise in the turbine room with a house generator attached to the generator end of each machine. They cover approximately the same space as each of the No. 1 and No. 2 triple units. All auxiliaries are electrically driven except the air pumps, which are of the steam-jet type. Electrically driven auxiliaries are essential for realizing the full advantage of main turbine

bleeding for heating boiler feedwater. The principal auxiliaries of the main unit are normally carried by the house generators, but they are also controlled by a device known as a " voltage chaser " which automatically throws over from the house-generator bus to the station-house bus, or vice versa, in case of the loss of voltage in either. This system has proved of great value in preventing a station outage when such loss of voltage has occurred.

14 With all units down, the station can be started from outside power, or in extreme emergency, with the outside power off, the

FIG. 2 INTERIOR OF COLFAX STATION, SHOWING FIRST AND SECOND INSTALLATIONS IN BACKGROUND AND THIRD INSTALLATION IN FOREGROUND

house turbines installed with Nos. 1 and 2 may be used for starting up.

15 Condensate is normally used as a cooling medium for the generator air coolers, transformers, and turbine-bearing oil, and to condense the steam from the steam-jet vacuum pump; the condensate flows from the hot well through the equipment in the order named. Under full-load conditions about 29 deg. fahr. is added to the condensate due to losses recovered. This, however, is not a net thermal recovery, as the reduction in steam extracted for the bleeder heaters reduces their efficiency about an equal amount. The advantage in using this cycle lies in the saving in the maintenance of the coolers and in the possibility of an even rate of heat transfer due to the clean water; also in the preclusion of chance of an outage due to clogging by leaves or by the scum

which prevails at times in the available raw-water supply. For emergency conditions a separate generator-air cooler section was installed for raw water. Up to the present time this emergency cooler has not been required for operation. The transformers operate on a water-to-water heat-exchanger system arranged so that raw water may be used as a cooling medium in emergency, without passing raw water through the transformer coils. The turbine-bearing-oil coolers are also provided with an emergency connection for the same purpose.

16 The boiler-feed pumps for the third-unit installation are driven by 500-hp., 50-deg.-continuous-rating, 2200-volt, 3-phase, 60-cycle, 1800 to 1630-r.p.m. variable-speed, Westinghouse motors with type "CW" rotor windings. The interesting feature of the installation is the new type of motor control used.

17 The control for each motor consists of a Hagan automatic pressure regulator in which a balanced piston is actuated by changes in the difference between the boiler-feedwater pressure and the steam pressure.

18 The piston rod of this regulator is attached to a rack actuating a gear, which in turn operates a notching drum. The notching drum closes a circuit which operates a pilot motor driving the main controller a given distance until a releasing device opens the motor circuit. In case the piston movement of the pressure regulator continues, the pilot motor continues to notch ahead until the increased speed of the motor causes sufficient pressure to build up to bring the regulator to rest.

19 The controller consists of a slate panel on which are mounted the necessary protective relays, the pressure regulator, a hand-wheel controlling the motor starting resistance, and six push buttons. Two push buttons, "start" and "stop," close or open the oil circuit breaker in the motor primary circuit. Two push buttons, "fast" and "slow," control the change in speed of the boiler-feed-pump motor when the automatic pressure regulator is not in use; and two push buttons, "automatic" and "manual," change the control from the regulator to manual control. With this scheme it is possible to operate this equipment at any given speed or to switch to automatic speed control. These pumps normally operate as part of the third-unit heat-balance system, drawing water from the low-pressure heater and discharging through the high-pressure heaters into the common feed system of the station. In an emergency, however, they may draw their supply from the boiler-feed supply of units Nos. 1 and 2.

20 The third-unit extension required additional intake capacity, and a screen house of sufficient capacity to take care of the third and fourth extensions has been constructed. Owing to the heavy deposits of water-soaked leaves which occur during the late fall and early spring, it was found advisable to increase the screen protection, and the new intake has been equipped with two rows

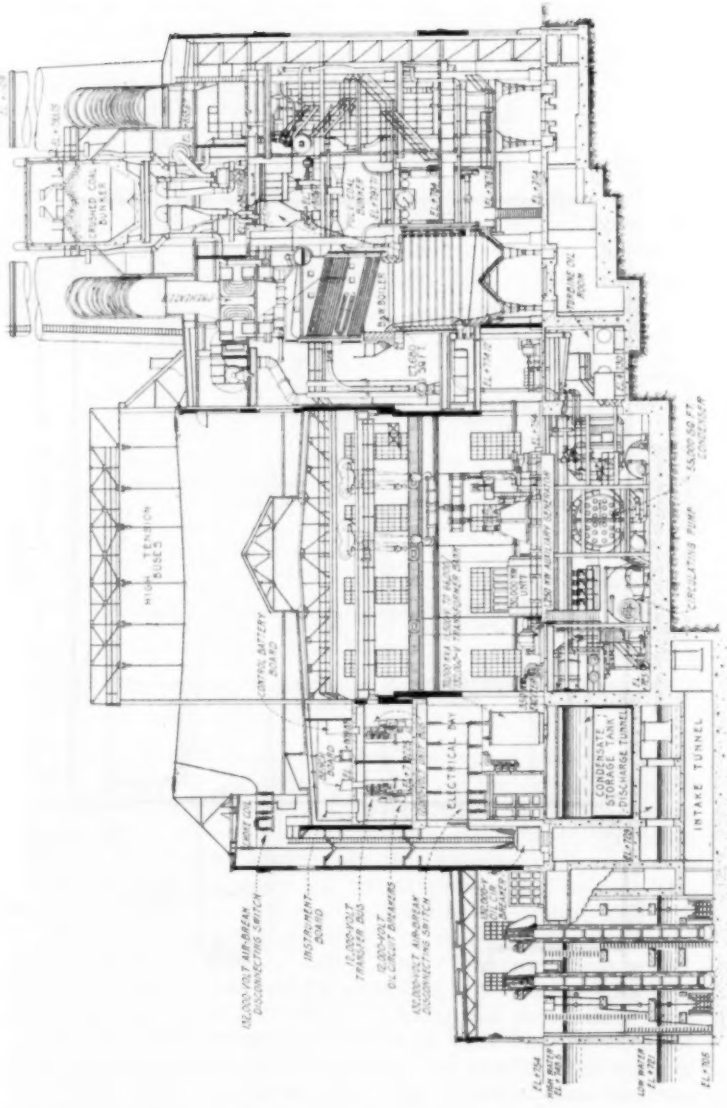

FIG. 3 CROSS-SECTION OF THE THIRD UNIT EXTENSION, COLFAX STATION

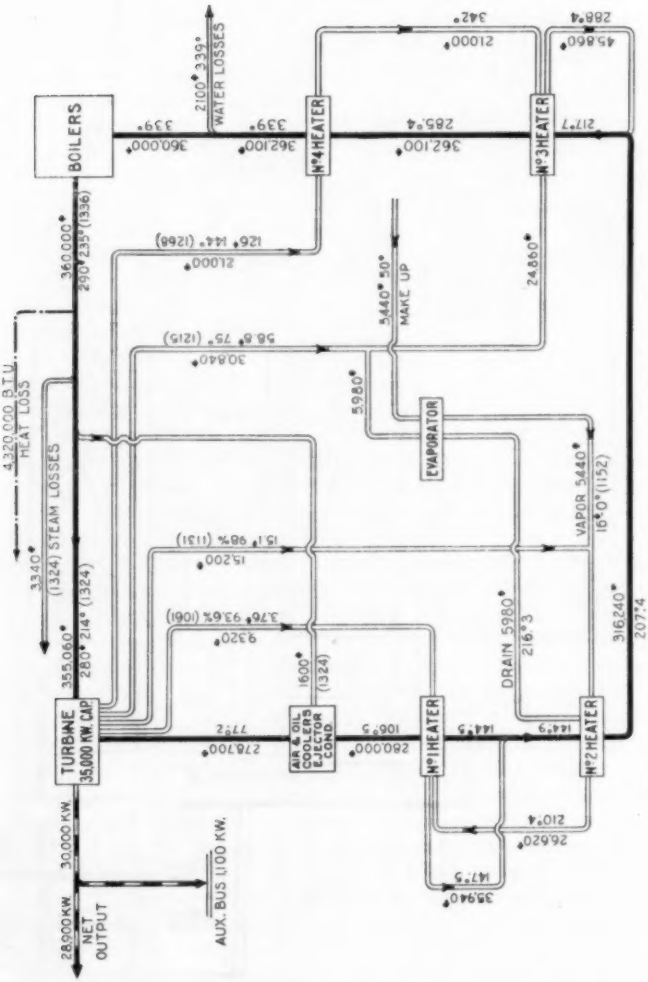

FIG. 4 HEAT BALANCE SYSTEM, THIRD-UNIT EXTENSION, COLFAX STATION

of screens, the front row having 1½-in. mesh and the rear row ⅜-in. mesh. This affords a double protection in the event that the front row becomes overloaded and fails. The discharge tunnel is a continuation of the Nos. 1 and 2 units tunnel.

21 Fig. 3 is a cross section of the third-unit extension and shows in detail the arrangement of the station.

22 Fig. 4 shows in detail the heat-balance system.

Boiler-Room Operating Results

23 The boiler-room operating results of the first unit were covered by the author in a paper entitled Boiler-Room Performance and Practice at Colfax Station,[1] presented to the Society at the Annual Meeting in December, 1922. These in general,

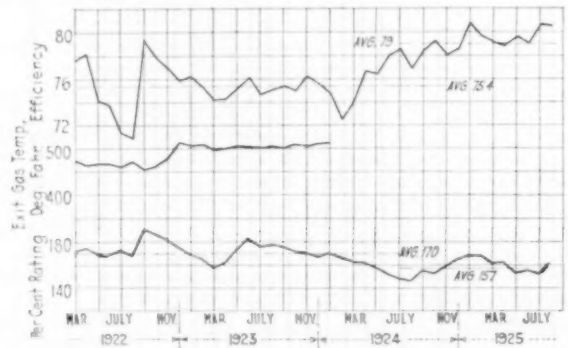

Fig. 5 Overall Operating Results, January, 1922, to August, 1925, Colfax Station

in so far as applies to the stoker-fired equipment, have been maintained practically the same. Fig. 5 gives the overall operating results from January, 1922, to August, 1925. The average exit gas temperatures are not given after the installation of preheaters because temperature readings on the preheater boilers are now taken above the preheater and the comparison is of no use. This curve, therefore, has not been continued from January, 1924.

24 Fig. 6 shows the availability of all boilers for the period from January 1, 1921, to July 1, 1925. The unavailable period, which averages 10 per cent for all boilers, is the time required for cleaning, repairs, and alterations. During this period forty-three tubes have been replaced; thirty-six of these were removed for test purposes to determine the extent and ascertain the cause of corrosion which was taking place. The results of these observations will be given to the Society at a later date. Four tubes have

[1] Trans. A.S.M.E., vol. 44 (1922), p. 217.

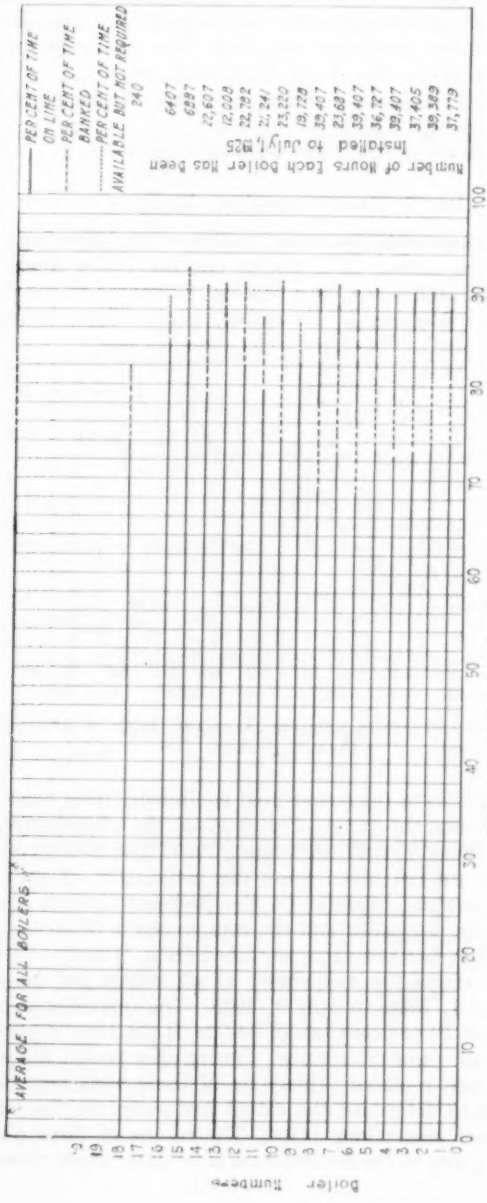

FIG. 6 AVAILABILITY OF ALL BOILERS, JANUARY 1, 1921, TO JULY 1, 1925

been removed due to cutting from soot-blower elements and three due to small cracks and holes which developed in the tubes.

Turbine-Room Operating Results

25 The overall operating results of the entire station from January, 1922, to August, 1925, are shown in Fig. 7. The high heat rate from May to September, 1922, was due to a coal strike which made it necessary to burn coals of widely different grades and characteristics. During the period of January, February, and April of 1923 and January and February of 1924, the units were operated with one or more of the elements out of service. This

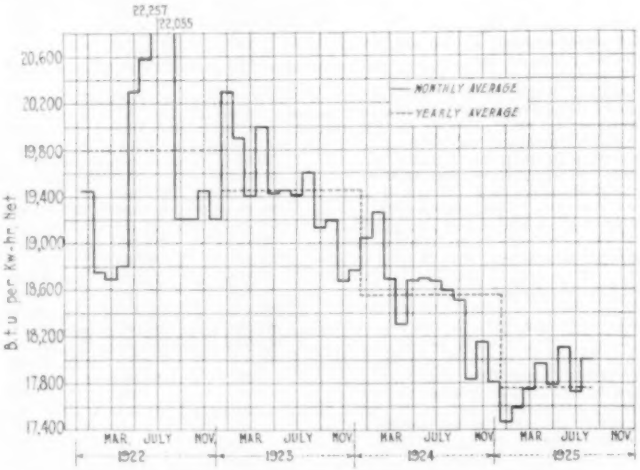

Fig. 7 Overall Operating Results, January, 1922, to August, 1925, Colfax Station

materially increases the heat rate of the turbine, the water rate rising from 11 to 18 or 20 lb. for such operation. The yearly average curve shows the effect of the installation of preheaters, pulverized-fuel system, and four-stage bleeding, which started in 1924 and was completed in 1925. As at present arranged, precise determination cannot be made of the heat consumption of the complete third-unit installation by itself. Tests have been made, however, which show the heat rate at the boiler nozzle for various loads to be:

Full load	12,752	B.t.u. per net kw-hr. delivered
Three-quarter load	13,021	" " " " "
Half load	14,200	" " " " "

The results of these tests are given in detail in Table 1.

TABLE 1 TEST OF 3-A ELEMENT, APRIL 24, 25, 1925, AT COLFAX STATION, AVERAGED READINGS

Date	April 25, 1925	April 25, 1925	April 24, 1925
Time	1 p. m. to 4 p. m.	9 a. m. to 12 a. m.	1.30 p. m. to 4.30 p. m.
Duration, hr.	3	3	3
Load, total, ind. kw	15,371	22,964	30,444
Load, main gen., ind. kw	14,967	22,552	30,016
Load, aux. gen., ind. kw	404	412	428
Load, total, int. kw	15,850	23,400	31,433
Load, main gen., int. kw	15,450	23,000	31,033
Load, aux. gen., int. kw	400	400	400
Power factor, per cent	69.0	82.0	87.3
Frequency, cycles	60.9	60.7	60.9
Voltage	11,967	12,040	11,970
Amperes A	1,031	1,298	1,637
" B	1,017	1,295	1,656
" C	1,020	1,275	1,620
Vacuum board, in. Hg	28.35	28.25	28.15
Back-pressure board, in. Hg abs.	0.92	1.04	1.17
Vacuum, cellar, in. Hg	28.53	28.55	28.52
Back pressure, cellar, in. Hg abs.	0.74	0.74	0.80
Absolute pressure, gage, in. Hg abs.	0.90	0.96	1.15
Exhaust temperature, deg. fahr.	77.8	79.1	85.2
Corresponding back pressure, in. Hg abs.	0.99	1.00	1.22
Hotwell temperature, deg. fahr.	76.	77.	83.3
Barometer reading, in. Hg	29.269	29.289	29.321
Barometer temperature, deg. fahr.	80.3	76.7	83.7
Hotwell discharge, ind., lb. per hr.	161,000	216,300	292,000
" " int., lb. per hr.	162,667	217,600	294,000
Water to No. 1 heater, ind., lb. per hr.	162,000	217,400	295,000
" " " " int., lb. per hr.	160,667	216,000	294,000
High pressure drains, ind., lb. per hr.	20,300	32,700	52,400
" " " int., lb. per hr.	20,333	32,300	49,000
Low pressure drains, ind., lb. per hr.	20,800	27,700	40,700
" " " int., lb. per hr.	22,333	31,700	42,700
Water to boilers, ind., lb. per hr.	204,000	272,800	370,500
" " " int., lb. per hr.	201,000	266,600	361,666
" " " chart, lb. per hr.	205,000	271,000	371,000
Line steam temperature, deg. fahr.	566.	598.	591.7
Throttle pressure, lb. per sq. in.	270	267	266
Primary pressure, lb. per sq. in.	181	254	265
Secondary pressure, lb. per sq. in.	62	90	240
Tertiary pressure, lb. per sq. in.	50	80	127
Heater No. 1 vacuum heater in. Hg	24.8	23.4	21.2
" " 2 vacuum heater in. Hg	12.3	5.4	2.2
" " 3 pressure heater, lb. per sq. in.	20.0	31.9	50.3
" " 4 pressure heater, lb. per sq. in.	55.0	81.3	123.0
Water to evap., ind. lb. per hr.	7,300	6,950	6,670
Water to evap., int. lb. per hr.	7,333	6,700	6,170
Evap. raw water inlet, temp., deg. fahr.	67.2	66.0	64.4
Inlet steam press. to evap., lb. per sq. in.	15.	19.2	28.9
Inlet steam temp. to evap., deg. fahr.	291.0	319.0	356.7
Evap. vapor vacuum, in. Hg	2.0	0.0 lb. per sq. in.	4.5
Evap. vapor temp., deg. fahr.	185.0	200.4	215.0
Evap. drain temp., deg. fahr.	213.0	216.3	230.3
Drain temp. heater No. 1, deg. fahr.	126.0	138.0	151.6
" " " " 2, deg. fahr.	175.0	190.3	208.0
" " " " 3, deg. fahr.	252.0	269.0	288.3
" " " " 4, deg. fahr.	256.0	278.0	300.7
" " " " 3, after cooler, d.fahr.	250.0	266.1	284.7
Steam temp., heater No. 1, deg. fahr.	129.0	141.2	153.2
" " " " 2, deg. fahr.	208.0	220.7	236.4
" " " " 3, deg. fahr.	291.0	323.9	370.0
" " " " 4, deg. fahr.	385.0	422.0	501.5
Cond. temp. out of heater No. 1, deg. fahr.	118.0	140.2	152.7
" " " " " " 2, deg. fahr.	184.0	198.0	213.4
" " " " " " 3, deg. fahr.	253.0	273.1	294.2
" " " " " " 4, deg. fahr.	303.0	325.0	348.9
" " in heater No. 2, deg. fahr.	127.0	143.8	155.0
" " " " " 3, deg. fahr.	210.0	214.0	221.3

Air ejector steam pressure, lb. per sq. in.	215.0	214.0	211.0
" " steam temp., deg. fahr.	535.0	568.4	559.8
" " cond. temp., in, deg. fahr.	96.0	94.0	96.0
" " cond. temp., out, deg. fahr.	110.0	104.2	104.1
" " raw water in, deg. fahr.	65.0	64.7	64.0
" " raw water out, deg. fahr.	68.0	66.2	66.9
" " primary drains temp., deg. fahr.	110.0	105.3	100.0
" " secondary drains temp., deg. fahr.	122.0	118.0	116.0
" " air discharge, cu. ft. per min.	13.8	8.0	5.9
" " secondary drains, lb. per hr.	1,528	1,362	1,380
Oil coolers water in, temp., deg. fahr.	91.7	90.2	93.5
" " water out, temp., deg. fahr.	96.0	93.9	96.0
" " oil in, temp., deg. fahr.	141.0	139.2	142.0
" " oil out, temp., deg. fahr.	112.0	109.0	111.6
Circulating water in, temp., deg. fahr.	62.0	61.1	58.6
" water out, temp., deg. fahr.	68.7	70.0	70.7
" pump disch. press., lb. per sq. in.	3.8	3.8	5.0
" pump suction press., in. Hg.	1.8	1.8	1.5
" pump r.p.m.	298.	297.3	298.
Mason regulator pressure, lb. per sq. in.	68.3	66.4	72.0
G. R. coolers, raw water in, temp., deg. fahr.	62.0	61.1	58.7
" " raw water out, temp., deg. fahr.	66.	65.9	63.0
" " transformer cooling water in, temp., deg. fahr.	91.9	92.7	86.0
" " transformer cooling water out, temp., deg. fahr.	64.0	62.2	61.1
U fin air coolers, air in, temp., deg. fahr.	131.0	135.0	143.0
" " " air out, temp., deg. fahr.	88.0	89.0	94.0
" " " cond. in, temp., deg. fahr.	76.7	78.0	84.0
" " " cond. out, temp., deg. fahr.	92.0	90.1	94.0
Gland water leak off, lb. per hr.	3,773	4,049	4,463
Gland water leak 2, temp., deg. fahr.	110	118	149
Governor oil pressure, lb. per sq. in.	50.8	50.9	50.5
Bearing oil pressure, lb. per sq. in.	6.9	6.8	6.5
Thrust bearing oil pressure, lb. per sq. in.	4.3	4.3	4.2
Thrust bearing oil temp., deg. fahr.	124	122	125
Bearing oil No. 1, deg. fahr.	133	133	134
" " Nos. 2 and 3 temp., deg. fahr.	154	153	154
" " No. 4 temp., deg. fahr.	142	139	143
" " No. 5 temp., deg. fahr.	112	108	114
Bearing oil to bearings temp., deg. fahr.	111	110	111.6
Hotwell pump discharge press., lb. per sq. in.	42.0	45.1	52.9
H. p. drain pump discharge press.,[1] lb. per sq. in.	32.0	36.0	38.0
L. p. drain pump discharge press.,[1] lb. per sq. in.	30.5	31.5	39.5
Gland water press., lb. per sq. in.	8.0	8.0	8.0
Turbine r.p.m.	1820	1820	1840

[1] This pressure was reduced for test period by speeding up boiler-feed pump in order to make these pumps handle the water.

26 As bearing upon an opinion more or less current that modern steam turbines and boilers of unusually large capacity are unreliable, a special study has been made to show time availability and outages on the units installed in Colfax Station. Fig. 8 shows the availability of all the turbo-generator units for the period during which they have been installed. The record is a creditable one having in mind the fact that units of great capacity have been in a distinct period of development since 1918. The availability of boilers is discussed in the paragraphs on boiler-room operating results. The following paragraphs contain some comment on availability of turbines.

27 The first three-element turbine has been installed since December, 1920, and up to July, 1925, has been available wholly or partially for 93 per cent of the time and as a complete unit 70 per cent of the time. The principal cause of the outages has

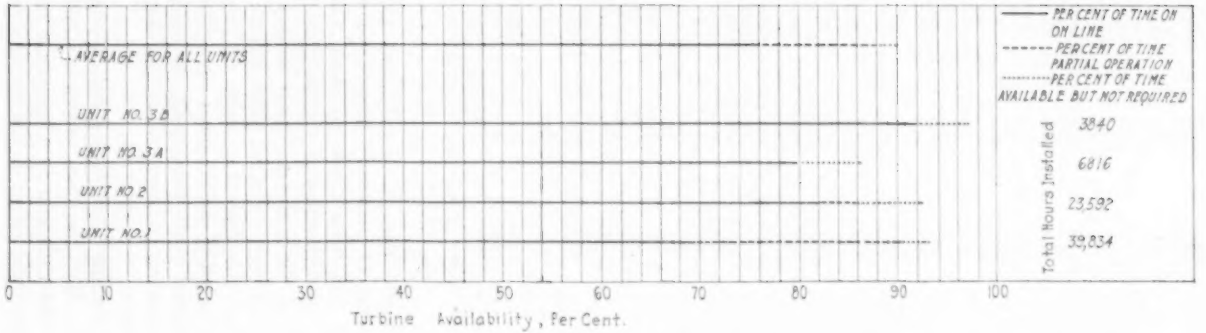

FIG. 8 AVAILABILITY OF ALL TURBO-GENERATOR UNITS FOR PERIODS FOR WHICH THEY HAVE BEEN INSTALLED, COLFAX STATION

been blade and dummy rubbing; the total outage due to this cause has been 1587 hours for the period. The remainder of the outage time has been for various minor repairs and changes.

28 The second three-element unit was installed in October, 1922, and up to July, 1925, has been available wholly or partially for 85.5 per cent of the time and as a complete unit for 82 per cent of the time. It has been out for repairs and alterations 1788 hours, or 7.5 per cent of the time, the principal cause of the outage on this machine being blade and dummy trouble. The total outage due to this cause has been 950 hours. The remainder of time out was for minor repairs and changes.

29 The third-unit installation has been in service less than a year. The first unit of this installation has been available 86 per cent of the time. It has been out for

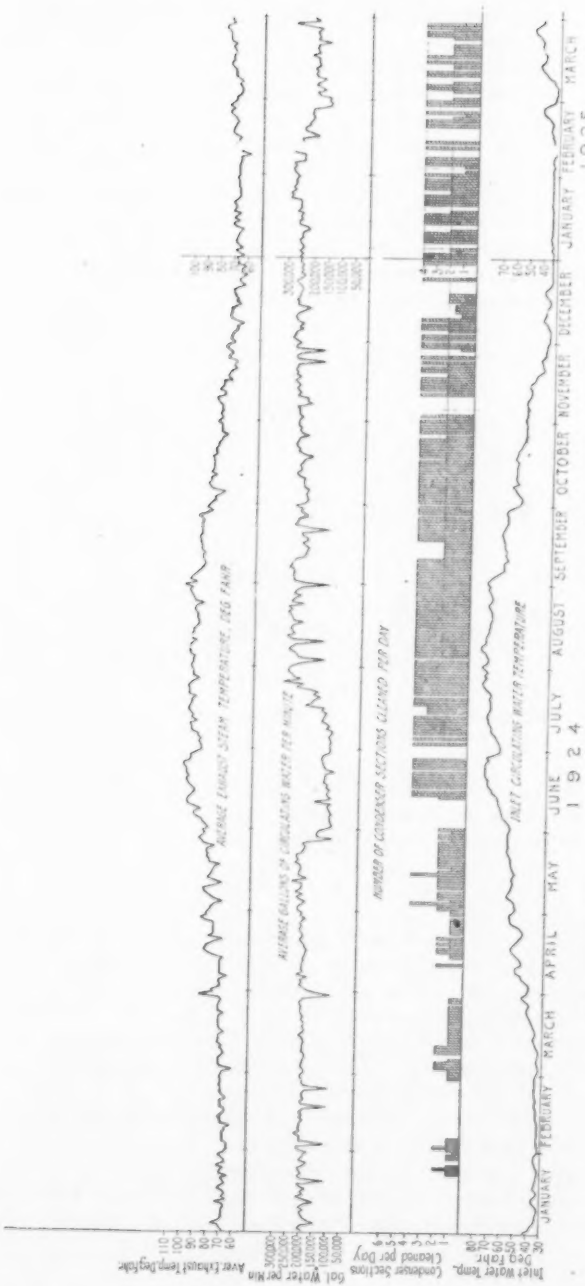
Aver. Exhaust Temp. Degree
Gal. Water per Min
Condenser Sections Cleaned per Day
Inlet Water Temp. Deg F.
AVERAGE EXHAUST STEAM TEMPERATURE, DEG. FAHR.
AVERAGE GALLONS OF CIRCULATING WATER PER MINUTE
NUMBER OF CONDENSER SECTIONS CLEANED PER DAY
INLET CIRCULATING WATER TEMPERATURE
JANUARY FEBRUARY MARCH APRIL MAY JUNE JULY AUGUST SEPTEMBER OCTOBER NOVEMBER DECEMBER JANUARY FEBRUARY MARCH
1924
1925

FIG. 9 FREQUENCY AND EFFECTIVENESS OF CONDENSER CLEANING OPERATION, COLFAX STATION

inspection and repairs 928 hours. The second part of the installation has been available 92 per cent of the time and out for inspection and minor changes 111 hours.

30 As previously mentioned, considerable difficulty has been encountered at this location on account of scum in the river, especially in the summer time, which has made it necessary to clean the condensers very frequently. In order to facilitate this operation, all condenser shells are equipped with arrangements for pumping the water from the water section overboard, so as to prevent water and debris from being deposited on the basement floor when the condensers are opened. Fig. 9 shows the frequency and effectiveness of the cleaning operation.

31 Fig. 10 shows the maximum and the average monthly temperatures of condensing water prevailing for the past four years.

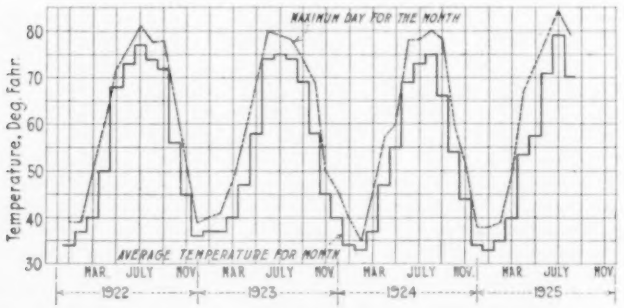

FIG. 10 MAXIMUM AND AVERAGE MONTHLY TEMPERATURES OF CONDENSING WATER, COLFAX STATION

It will be noted that there has been a gradual rise in the temperature during the summer months, and that a mean of 84 deg. fahr. was reached during July of this year.

COSTS

32 The proper objective in power-station design is the lowest cost of electric energy inclusive of fixed charges as well as operating expenses. If this objective is followed, it is reasonable to expect differences in investment in different localities. Investment may also be influenced considerably by the physical conditions to be met at different sites, such as foundation difficulties, length of intake or discharge tunnels, etc. Moreover, there is divergence of practice in just what is included in or omitted from the total cost figures which are published and discussed.

33 These various influences doubtlessly explain a part of the very wide differences in costs discussed at the present time, varying from seventy dollars to more than two hundred dollars per

kilowatt. They do not, however, explain all the differences, since the capabilities of the designers and the builders are important factors in the final cost. A consideration of the details is therefore essential before drawing conclusions as between one design and another unless the factors indicated above are known to be similar. It has not been customary to publish such details of power-station costs, and the author found it difficult to obtain details outside of

FIG. 11 COST DIAGRAM, COLFAX STATION, CAPACITY 190,000 KW.

his own organization. Although no details are given in this paper, the author would be glad to exchange such detail costs with any one interested.

34 The totals given below include all costs for the power station, sea wall, intake and discharge tunnels, outside switching station, and all overhead costs, such as engineering and construction fees.

35 The cost of the Colfax Station at the completion of the second unit was $13,982,321.81, or $116.51 per kw. for an installed capacity of 120,000 kw. The third-unit installation, of 70,000-kw.

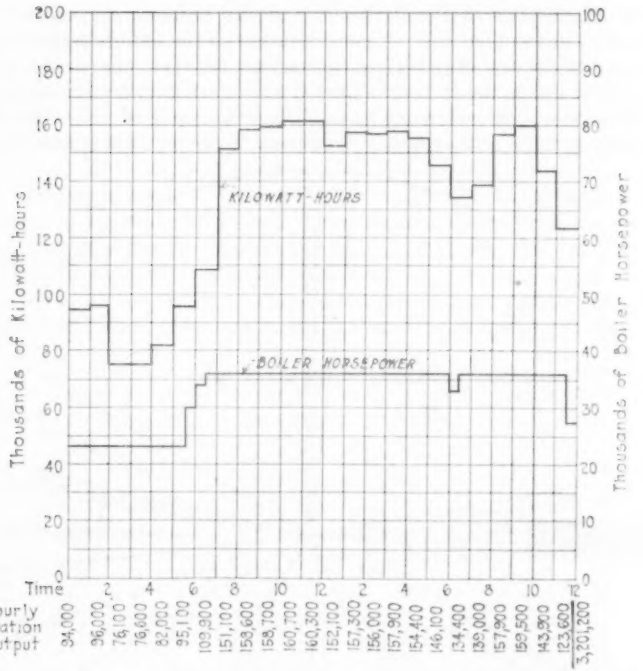

FIG. 12 TYPICAL DAILY LOAD CURVE, TUESDAY, SEPTEMBER 15, 1925, COLFAX STATION

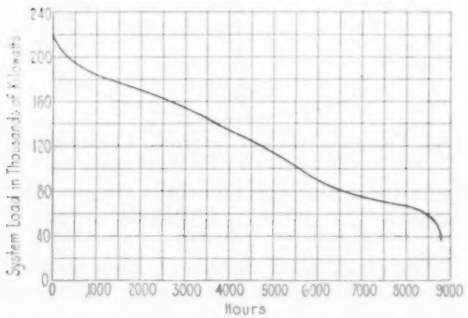

FIG. 13 LOAD-DURATION CURVE FOR ENTIRE DUQUESNE LIGHT COMPANY SYSTEM FOR THE YEAR 1924

capacity, cost $8,019,483, making a total cost to date of $22,001,-804.81, or $115.80 per kw. for an installed capacity of 190,000 kw. This cost includes building, foundations, reserve boilers, and service equipment which are applicable to the fourth unit.

36 At the present time the Duquesne Light Company have under way the installation of a fourth unit of 80,000-kw. capacity at an estimated cost of $7,025,000, which would give a total installed capacity of 270,000 kw. at a total cost of $29,026,804, or $107.50 per kw. These figures are exclusive of an outside coal-handling plant and a coal railway to the mine, constructed at a total cost of $422,100.32. It will be noted that the unit cost decreases as plant additions are made due to the necessity of constructing building and installing equipment which also serve the future additions.

37 Fig. 11 is a cost diagram for the 190,000-kw. development.

38 Fig. 12 shows a typical daily load curve and boiler-capacity curve.

39 Fig. 13 is the load duration curve for the entire Duquesne Light Company system for the year 1924.

Survey of Results

40 The test results indicate the following performance:

Output (net), kw	30,333	22,400	15,050
Heat rate (net), B.t.u. per kw-hr	12,752	13,021	14,200
Water rate (net), lb. per kw-hr	12.84	12.63	13.63
Line steam pressure, lb. per sq. in	266	267	270
Line steam superheat, deg. fahr	180	186	153
Exhaust pressure, in. Hg	1.22	1.00	0.99
Steam from Bleeder No. 1, lb. per hr	12,666	6,556	322
" " " No. 2, lb. per hr	14,344	7,380	4,108
" " " No. 3, lb. per hr	35,013	24,065	17,230
" " " No. 4, lb. per hr	20,207	14,193	10,270
Total bled, lb. per hr	82,230	52,194	31,930

DISCUSSION

F. O. Ellenwood.[1] It seems to the writer that the author would have made his paper more valuable than it is had he given the additional cost that was incurred by the installation of heaters in 1925. The paper would then give a direct comparison of the relation of the reduced expenditure for fuel as compared with an additional investment. It also is to be regretted that he did not make clear in the tabulation of the data as to whether the pressures are absolute or gage.

The writer is inclined to question the statements in Par. 15 relating to the net recovery of heat losses by using the condensate as a cooling medium for the generator air coolers, transformers, and turbine-bearing oil, and to condense the steam from the steam-jet vacuum pump.

[1] Professor of Heat Power Engineering, Cornell University, Ithaca, N. Y. Mem. A.S.M.E.

Concerning Fig. 4 supposedly showing in detail the heat-balance system, there is the important question as to whether this shows the system as the designers planned it, or as it was obtained by test. If the latter, why does it not agree more closely with the test results given in Par. 40? To the writer there seems to be a large number of discrepancies in the thermal data.

H. W. LEITCH.[1] The extension of Colfax is of interest on account of its heat-balance system, as given in the paper, but is of particular interest on account of the opportunity which is presented of comparing the general performance of a modern powdered-fuel installation with a comparatively modern stoker installation. It is unfortunate, therefore, that the detailed test figures on the third section give results only as of the boiler nozzles. In other words, the test resolves itself into a test of turbines and heat balance. The overall load B.t.u. per net kw-hr. delivered under conditions of full load of 12,752 B.t.u., although only for three hours, shows remarkable heat recovery and good turbine performance. Assuming an overall boiler efficiency of 85 per cent as against approximately 80 per cent given as a maximum in Fig. 5, gives 15,000 B.t.u. input per kw-hr. output, or 16,000 B.t.u. at 80 per cent boiler efficiency.

It might be of interest as a check against this to note that a recent 24-hr. run was made on one Hell Gate section consisting of a 35,000-kw. turbo-generator and three 1590-hp. boilers equipped with economizers, superheaters, and water-tube side walls, all installed in 1923, and during this period the unit generated 639,000 kw-hr. with a weighed coal consumption of 643,000 lb., representing a consumption of 14,080 B.t.u. per kw-hr. generated. This run was made under conditions of 66-deg. fahr. circulating-water temperature, 8 deg. higher than at Colfax. It is expected that a run of this sort under winter conditions of low auxiliary power and high vacuum would indicate an increased efficiency to about cover a proportionate station loss, and the result would indicate the production of a kilowatt of net station output on less than a pound of good coal.

JOSEPH S. BENNETT.[2] Almost invariably when a plant with both stoker and pulverized-coal installations is discussed, a comparison is made between the results obtained with the two systems. As a rule the stoker is an obsolete type, and the conclusion reached is that stoker firing is quite an inferior system as compared to the modern pulverized-coal system installed. For example, in Fig. 5 there is a sharp drop in the efficiency when poorer grades of coal

[1] General Superintendent of Power Plants, United Electric Light & Power Co., New York, N. Y. Mem. A.S.M.E.

[2] Assistant Engineer, American Engineering Co., Philadelphia, Pa. Assoc-Mem. A.S.M.E.

were used due to the coal strike. The inference is that a variation in the coal quality would very seriously affect the operating results of a stoker-fired plant. It would seem that in making a fair comparison a stoker with proper equipment for burning the lowest-grade coals to be encountered should be used in the comparison; for example, it is quite probable that the lower grades of coal burned during the strike were of a character that would cause side-wall and bridgewall clinker trouble which would not have occurred had the equipment been designed to burn this poor grade of coal. Also, the ashpit in the plant we are considering is a shallow one, which would naturally show a fairly rapid increase in the ashpit loss when inferior grades of coal are burned.

With a properly designed clinker pit, which means a deep ash pocket with proper means for controlling the air supply and with bridgewall and side-wall protection to prevent clinker adhesion, the combustible in the ash ordinarily gives a loss of from $\frac{1}{3}$ to $\frac{1}{2}$ of 1 per cent of the total heat value of the coal fired. This result may seem rather extraordinary to those who have not been in touch with the more recent developments in stoker design. For example, the more recent design of ashpit has not only reduced the ashpit losses but also acts as a stabilizer, making good operation better and tending to offset poor operation. As a specific example of this, in one installation on which extensive tests were run quite recently with an ashpit of the type commonly used for the last three years, certain efficiencies were obtained. Later, under practically the same conditions, a series of tests were run with the same size stoker equipped with the latest type of ashpit. The efficiencies at from 150 to 200 per cent of boiler rating were practically the same for both stokers. At 300 per cent of boiler rating, the later type of stoker showed an increase in efficiency of about 6 per cent, while at 400 per cent the gain amounted to from 8 to 10 per cent. The efficiency at 400 per cent of boiler rating on the later type stoker was 78 per cent for the boiler alone without an economizer or preheater.

The higher efficiency is not, of course, due solely to the ash pit but to the improvement in combustion conditions throughout the whole length of the stoker.

R. J. S. Pigott.[1] This paper is particularly valuable as showing the tremendous effect of the multiple-bleed-point regenerator system on modern turbines. The tremendous decrease in B.t.u. rate is due to the extremely new economical section. The effect of this, of course, will increase as the amount of kilowatt-hours relative to the total amount of kilowatt-hours increases, ultimately reaching the B.t.u. rate of the new section.

A station very similar to that explained and described by the author is the one in Toronto, running since April, 1925. The

[1] Public Service Production Co., Newark, N. J. Mem. A.S.M.E.

physical layout is to a large extent similar to the new sections of the Colfax Station. While test results are not yet available, sufficient data are on hand to indicate a net B.t.u. rate of something under 15,000, although these optimistic figures were not used in computing the probable cost. At the present time the station is using 16,000 net B.t.u. per kilowatt-hour and the rate is continually going down. In this particular installation the rate of driving the boilers was carried a little higher than in the Colfax station and other powdered-coal plants.

The first layout was designed for a maximum boiler operation of 300 per cent of rating for the load turbines, with the possibility of being forced to 400 per cent of rating at times of boiler outage, figuring on one boiler out of four. This condition would of course apply to the first two units installed. It would be lowered as the number of units in the station is increased. At this high load, the plant gave trouble due to excessive smoking. In the plant in which this occurred the water screens were used to a larger extent than ever before.

The question of air distribution comes up in the handling of a powdered-coal fire. The mere large volume of these furnaces is not sufficiently reasonable to give good combustion without smoke unless the most careful attention is given to the control of the fires. The writer believes that it would not have been possible to operate this plant satisfactorily without combustion control. During the period when hand control was used for running this plant, excessive trouble from smoking occurred. It was found, however, that a great deal of this smoke could be avoided by studying the air supply. It is true that above 250 per cent of rating smoking has not yet been eliminated, even with automatic control, although there is no reason to suspect that the station will not achieve smokeless combustion. It seems to be necessary to furnish more primary air in order to shorten the flame. In any case the flame is to be kept away from the boiler surfaces if smokeless combustion is to be attained.

The comparison of heat balances is becoming a difficult matter. No two stations built have the same heating arrangements. In due course we shall have to make a study to ascertain which arrangement is the most efficient and stop changing heat-balance systems. Of course, the engineers followed their own ideas in the design of this plant, and so far it has worked out fairly well. The attempt was made to raise the feedwater temperature by bleeding, using such means as would not cramp the bleeding cycle with the temperature at 380 deg. fahr. At the present time it is only carried to 360 deg. fahr. due to troubles caused by casing expansion on the boiler-feed pumps. Most of the boiler-feed pumps are not safe for higher temperatures than 300 deg. fahr. If the temperatures are higher, the pumps should be redesigned for safety. The feedwater temperature is constant within 5 deg. fahr. or more without

any hand manipulation, so it is evidently possible to get a heat-balance system automatic without adjustment.

A. K. Bak.[1] In Par. 5 the author states that while the operating pressure was kept the same in the third-unit extension the steam temperature was increased from 600 to 650 deg. fahr. As several other stations have gone to from 700 to 725 deg. fahr. it would be of interest to hear the reason for not going higher than 650 deg. fahr. Incidentally it might be noted that the tests reported in Table 1 show steam temperatures below 600 deg. fahr. for all tests.

In the same paragraph it is also stated that it was decided to install two turbine units instead of one larger unit. The original installation contains 60,000-kw. three-element units, whereas the new machines apparently are 35,000-kw. single-barrel units. In view of the general trend toward larger units, what is the reason for this change?

It is interesting to note that whereas the side walls in the furnaces are protected either by screen tubes or by air cooling, the furnace bottoms have both, and one wonders whether or not the added complication of hollow bottoms and air cooling pays for itself.

In Par. 25 the statement is made that the high heat rate from May to September, 1922, was due to burning coal of widely different grades and characteristics. Using the curves on Figs. 5 and 7, and comparing the averages for these four months with the yearly averages for 1922, one finds that whereas the boiler efficiency was about 2.9 per cent lower, the B.t.u. per kw-hr. net increased some 7.5 per cent, and for the two months of July and August the boiler efficiency was about 4 per cent lower than the yearly average, but the B.t.u. per kw-hr. net increased almost 12 per cent. Although the auxiliary power consumption undoubtedly would be greater during the above-mentioned period, it would hardly account for the difference, and it appears that there must have been some other reason besides the widely different grades of coal.

Table 1 gives the results of three tests on one of the units. A closer study of this table reveals some very surprising information indeed. The back pressure at the board is from 0.2 to 0.37 in. Hg higher than that observed in the cellar. It is not stated where in the condenser the pressures were measured, but one must assume that the difference indicates the drop through the condenser, which seems very high. Apparently the same barometer reading was used in determining the two back pressures without taking into account the difference in elevation.

The evaporator vapor temperature is lower than that corresponding to vapor pressure; in other words, the evaporator produces supercooled vapor. In fact, the temperatures and pressures

[1] Municipal Power Plants, Copenhagen, Denmark. Mem. A.S.M.E.

as recorded check very poorly. As another example it might be mentioned that heater 3 shows a higher drain temperature than that corresponding to the pressure in the heater. With heaters one usually finds a certain relation between terminal difference and load, and the readings here presented show no consistency whatever. It is interesting to note that the last heaters receive steam of high superheat (up to 200 deg. fahr. superheated) and that the water is heated to a higher temperature than that corresponding to the pressure in the heater.

On load and water measurements the author reports both indicator and integrator readings, but without stating which one has been used in the calculations.

THE AUTHOR. Referring to Professor Ellenwood's question as to the cost of bleeder heaters, it is not possible to pick out any one item in a large development such as Colfax Station, except for comparative purposes, and say that it would or would not pay. The installation of bleeder heaters in a heat cycle immediately makes necessary, if proper economies and reliability are to be secured, the consideration of other means of auxiliary power supply; and if auxiliary shaft-driven generators such as at Colfax are used, the actual cost of the turbo-generator unit is increased, but there is a greater decrease in cost over the old auxiliary power situation made possible by the omission of house turbines, etc.

In the case of the Colfax four-stage bleeder system, at 50 per cent load factor, the return on the investment will be somewhere around 25 per cent on the actual cost of the bleeder system.

The pressures given in the heat-balance diagrams are absolute. The difference between the calculated figures on the diagram and the test figures give a good comparison of the accuracy with which the designers were able to predict the functioning of this system.

As to the question on the 29 deg. fahr., which were added to the hotwell water by the generator air coolers, oil coolers, and transformers, the addition of this heat to the cycle reduces the amount of steam which can be bled at the first bleed point by an amount equal to 29 deg. elevation of the condensate. This in turn will show a corresponding increase in condenser loss due to this 29 deg. addition of temperature in the condensate. It would have been better for thermal efficiency not to have had this temperature added to the condensate, but the safe and reliable operation of apparatus secured by providing clean cooling water at these points overbalances any efficiency which might be lost.

With reference to Mr. Leitch's comments, it should be noted that the average efficiency figures given cover only two of the new pulverized-fuel boilers with fourteen stoker-fired boilers. Two of the stoker-fired and the two pulverized fuel-fired boilers are equipped with air heaters. The actual efficiency of the new pul-

verized-fuel boiler plant is considerably higher than that indicated by the average figure.

The author realizes the difficulty in comparing power-station costs, as pointed out by Mr. Leitch. The costs given in the paper cover all costs with the exception of land.

The question was raised as to the reason for the variation of feed temperature between 300 and 350 deg. at varying load. A bleeder heater system is practically automatic, the work being done by the induction of steam into the heater system, and the steam flow into the various heaters naturally depends on the pressure condition existing at the bleeder points in the turbine, which in turn is affected by load.

In reply to Mr. Pigott's comments, the author's paper on heat balance given two years ago stated that Colfax had a boiler-nozzle heat rate of 14,200 B.t.u. If that be compared with the present boiler nozzle heat rate of 12,752 B.t.u., the difference will be the improvement due primarily to the different heat cycle.

Referring to the regulation of the pulverized-fuel plant, we are having some difficulties along the lines mentioned by Mr. Pigott, this being evidenced by smoke at times of improper air distribution, but we feel that experience will straighten out this matter. As far as primary air supply for the burners is concerned, we are arranging to use more air at this point and should not be surprised if we ultimately used as much as 40 per cent of the total air at the burners.

Referring to Mr. Bak's comments, the steam temperature was raised to a maximum of 650 deg. because the older turbines in the station would not safely operate at a much greater temperature. Further, the low temperatures in the tests, shown in Table 1, are due to the small number of boilers operating with the larger superheaters as compared with the other low-temperature superheaters in the station.

As to the reason for changing from the three-element 60,000-kw. unit to the two 35,000-kw. single-barrel turbines, this was entirely a commercial matter with the manufacturer. The new design of turbine was thought better than the old double unit.

As to the combination of water screens and air-cooled furnace bottoms, we are entirely satisfied by experience that they are justified.

In reference to the high heat rate from May to September, 1922, Mr. Bak has evidently neglected to take into consideration the fact that Fig. 7 reflects the operation of the entire station, while Fig. 5 shows boiler-room operation only. During this particular period, the turbine-room efficiency was also low. It would be difficult to account for all of these factors except in a general way; the effort to maintain operation under extreme difficulties probably lowered the morale of the operating forces and affected efficiencies throughout the station.

The results given in Table 1 were obtained from data taken from the regular station instruments which were calibrated to an accuracy of plus or minus 1 per cent. The difference in elevation of points at which condenser pressures were taken was approximately 25 ft. No attempt was made to correct for this small difference. The mercury column at the board measures pressure at the exhaust nozzle of the turbine and the cellar pressures were taken from the condenser hotwell. Considering the size of the unit, the difference in pressures indicated are not excessive, particularly as they represent differences in air content as well as friction losses.

We have found that the point at which the evaporator vapor pressures were taken give inaccurate results and they have not been considered in our calculations; the vapor temperatures, however, are correct.

We cannot check Mr. Bak's statements regarding heater temperatures. According to the Marks and Davis tables, which we have used in our calculations, the drain temperatures of both No. 3 and No. 4 heaters are lower than the corresponding temperatures of saturated steam in the heaters. This is due to the cooling effect of condensate in the tubes of the first pass of the heater over which the condensed steam passes before it reaches the heater hotwell. The temperature of water leaving the No. 4 heater corresponds very closely to the saturated temperature of the steam in the heater at all loads. Integrator readings have been used for the unit calculations made in Table 1.

No. 1987

STEAM BLEEDING AND TURBINE PERFORMANCE

TESTS SHOWING EFFECT OF STEAM BLEEDING ON TOTAL POWER-PLANT EFFICIENCY, ALSO EFFECT OF CHANGES IN VACUUM AND SUPERHEAT ON TURBINE AND CONDENSER PERFORMANCE

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The primary object of this paper is to give the results of tests which show the improvement in overall steam-power-plant economy due to single-stage steam bleeding. Detailed results of the tests are also described which show the changes due to steam bleeding in boiler-unit efficiency, degree of superheat, and the performance of the turbine-condenser unit.

In order to complete the study of the effects of steam bleeding it was necessary to show the effects of changes in superheat and condenser back pressure on the turbine performance. The results of these tests are given in the second part of the paper and are of interest as a separate study of steam-turbine performance. With this in mind, additional data such as the effect of air in the turbine exhaust steam and effect of changes in initial steam pressure are also given. The change in superheat covered the range from 100 to 300 deg. fahr. and the condenser back pressure was of sufficiently low value to show the economical value for loads above 25,000 kw.

The tests were made at the Lake Shore Station of the Cleveland Electric Illuminating Co. and the turbine tested was a Curtis-type, 17-stage machine of 30,000 kw. capacity. Steam was bled from the 13th stage.

THE test data outlined in the following are divided into two parts, the first of which is concerned with the net effect on steam-power-plant efficiency of single-stage steam bleeding, and the second of which is concerned with the characteristics of steam-

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turbine performance as affected by changes in condenser vacuum and steam superheat. The second part of the paper was felt to be a necessary supplement to the first since a change in the amount of steam bleeding is followed by changes in steam superheat and

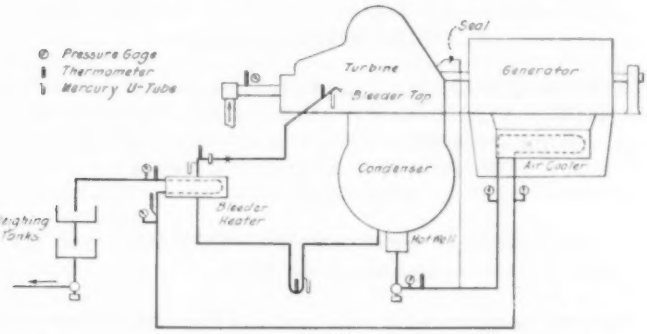


FIG. 1 INSTRUMENT AND PIPING LAYOUT USED IN TESTS

condenser vacuum, and it was felt that a summary of the results of steam bleeding which showed changes in steam temperature and condenser back pressure would be incomplete without showing the degree to which these changes affected the net result. These

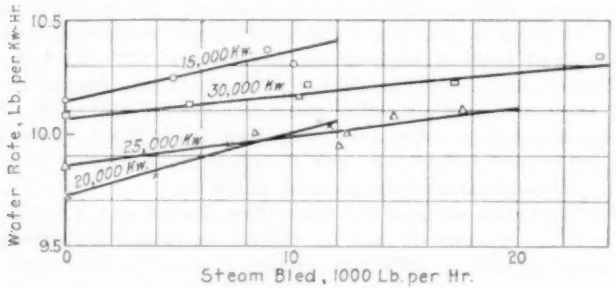


FIG. 2 EFFECT OF STEAM BLEEDING ON TURBINE WATER RATE
(Superheat, 250 deg. fahr.)

data will also be found of interest as a separate study of steam-turbine characteristics, and in this connection it is worthy of special note that the condenser back pressure was of sufficiently low amount as to indicate the economical vacuum for the turbine for loads above 25,000 kw. In order to furnish data for a complete study of turbine performance, additional test results are included in the second part of the paper which have no direct bearing on the subject of steam bleeding.

2 As this paper is largely of a research nature and of a comparatively new phase of power-plant performance, no attempt has been made to discuss the data from a standardization view-

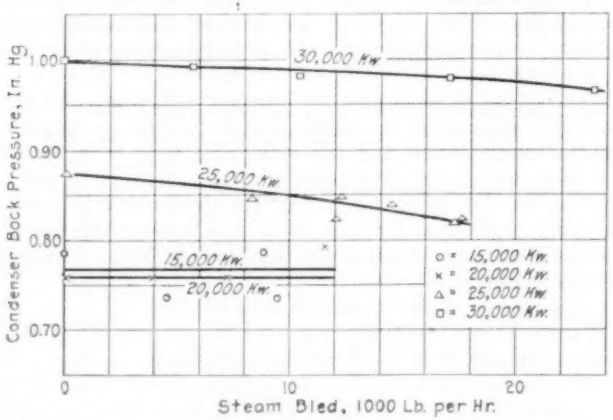


FIG. 3 EFFECT OF STEAM BLEEDING ON CONDENSER BACK PRESSURE

point. In order that the reader may have the proper perspective in reading this paper it is well to state that the object of these tests was not to develop something new but to furnish performance

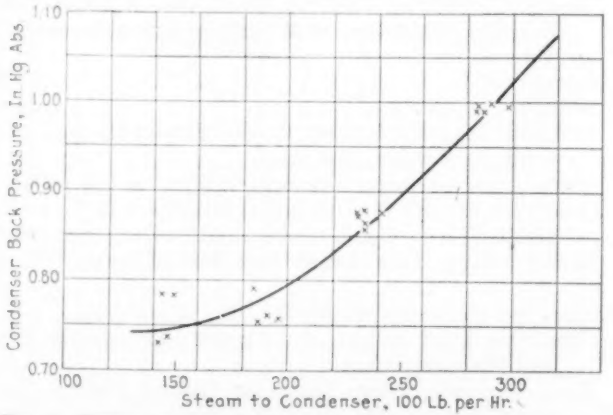


FIG. 4 EFFECT OF THE AMOUNT OF STEAM CONDENSED ON THE CONDENSER BACK PRESSURE
(Circulating water temperature 54.2 deg. fahr.)

data to those interested in the subject of steam bleeding. Emphasis is placed on the effect of steam bleeding on boiler-room performance.

PART I

EFFECT OF STEAM BLEEDING ON EFFICIENCY

3 In addition to the net effect on steam-power-plant efficiency of single-stage steam bleeding, the detailed changes on turbine, condenser, and boiler-unit performances as effected by steam bleeding are also given.

4 The phase of boiler-unit performance with which this paper is mainly concerned is the change brought about by changes in feedwater temperature, and the boiler-unit tests were therefore made with that object. The relation of the efficiency to the rate of driving of the boiler equipment was determined in previous tests, which were given by John Wolff in a paper which was published in *Mechanical Engineering* for January, 1925. This curve is shown in Fig. 11.

5 A summary of the equipment is as follows:

Boilers, Duplex Stirling, sq. ft. 30,600
Superheaters, B. & W. convection type, sq. ft. 4,520
Water-tube screen, sq. ft. 835
Economizers, Power Specialty Co., sq. ft. 22,080
Furnace volume above screen, cu. ft. 26,000
Turbine, General Electric Co., single-cylinder, 17-stage, 30,000-kw., bleeding, 13th stage
Throttle pressure, lb. per sq. in., gage 235
Steam superheat, deg. fahr 250
Generator, General Electric Co., type A.T.B.-4-30,000; capacity, 30,000 kw.; at 85 per cent power factor, 35,300 kva.
R.p.m. 1,800
Voltage 11,430
Ventilation, closed air system, General Electric fin-type cooler, condensate water, two-pass.
Exciter, direct-connected, capacity 210 kw. at 250 volts.
Condenser, Worthington Pump and Machinery Corp., surface, sq. ft. 53,000
Bleeder heater, Ross Heater and Manufacturing Co., sq. ft. 985

Turbine and Condenser Auxiliaries

Circulating-water pumps, two per unit, Worthington Pump and Machinery Corp.
Circulating-water pump drive, one General Electric Curtis steam turbine, 200 hp.; one General Electric induction motor, 200 hp.
Steam-jet vacuum pump, Worthington Pump and Machinery Corp., size, 3.20 cu. ft.
Hurling-water pump, Worthington Pump and Machinery Corp., size, 8. C. L., motor drive, General Electric induction motor, 75 hp.

Condensate pumps, two per unit, Worthington Pump and Machinery Corp., size 6.

Condensate-pump drive, one General Electric Curtis steam turbine, 60 hp., one General Electric induction motor, 60 hp.

Method of Testing

6 For all the turbine tests the piping and instrument locations were as shown in Fig. 1. The weighing tanks were calibrated with

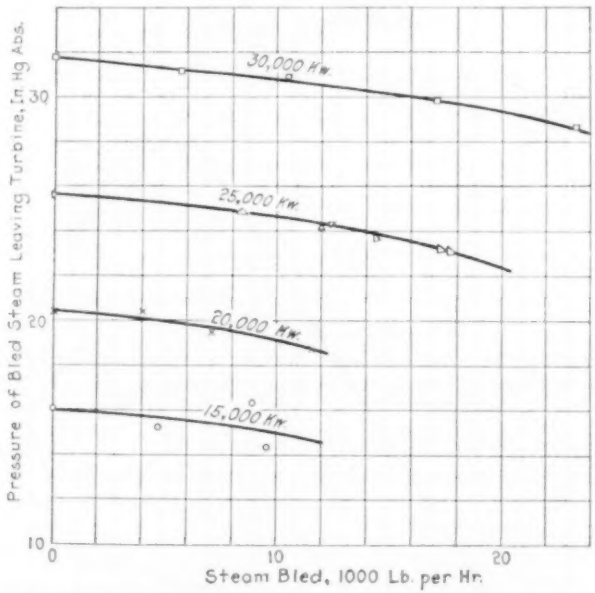


Fig. 5 Effect of Increasing Steam Bled on Pressure of Bled Steam Leaving Turbine

standard weights before the tests and were checked at different times during the tests. The kilowatt-hours were measured by rotating standards which were checked at the company's laboratory before the tests. All thermometers were compared at the operating range with thermometers which had been calibrated at the Bureau of Standards. All turbine tests were of one hour's duration, except those of 15,000 kw. or less, which were of two hours' duration. A period of from fifteen minutes to one hour under test conditions always preceded each test to insure constant temperature conditions.

7 To determine the effect of steam bleeding on the turbine characteristics, the turbine was tested at loads of 15,000, 20,000, 25,000, and 30,000 kw., with amounts of steam bled ranging from

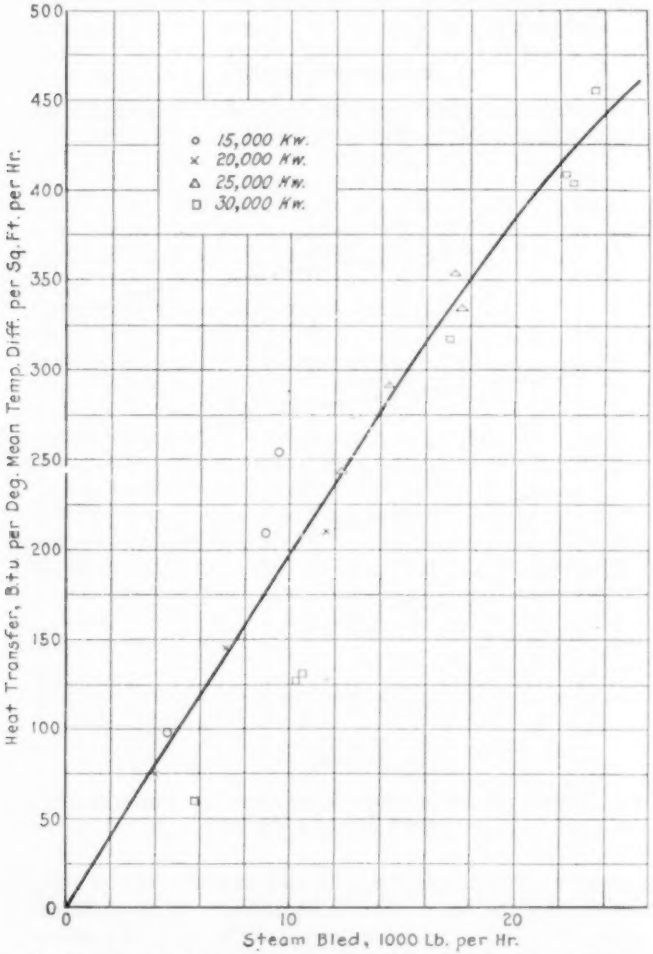


FIG. 6 EFFECT OF STEAM BLEEDING ON HEAT TRANSFER IN BLEEDER HEATER

zero to the maximum for each load. The amount of steam bled was calculated from a heat balance on the bleeder heater.

8 A similar method was used to determine the turbine characteristics under variable vacuum and superheat conditions with no steam bleeding.

9 To determine the effect of steam bleeding on the boiler-unit efficiency, the boiler was operated at constant rate of driving and

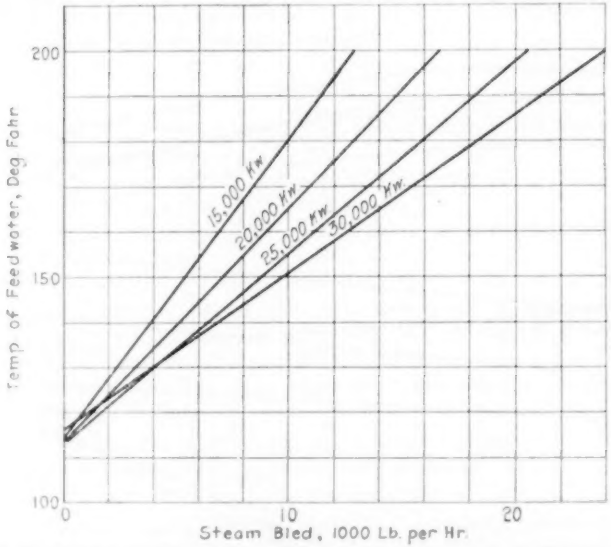


FIG. 7 EFFECT OF STEAM BLEEDING ON TEMPERATURE OF FEEDWATER
(These curves are representative of the actual operating cycle and include the temperature rise due to the steam auxiliaries.)

excess air and tested at about 100, 150, and 200 deg. fahr. feedwater temperatures. Three sets of these tests were run in order to get an average.

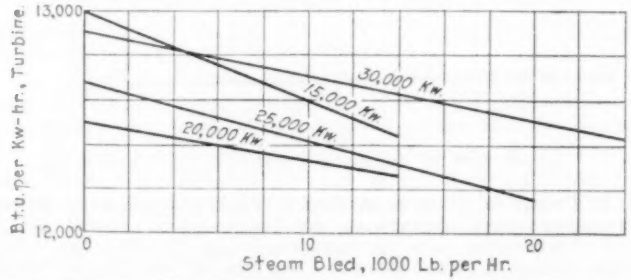


FIG. 8 EFFECT OF STEAM BLEEDING ON B.T.U. PER KW-HR. CHARGEABLE TO TURBINE UNIT ALONE

10 Since the boiler temperatures gradually rose during operation between soot-blowing periods, and since each set of three tests were run with no soot blowing during the period of each set

of tests, the order of the tests with respect to feedwater temperature on the first set was 200, 100, and 150 degrees, on the second 100, 150, and 200, and on the third 200, 150, and 100. Each test

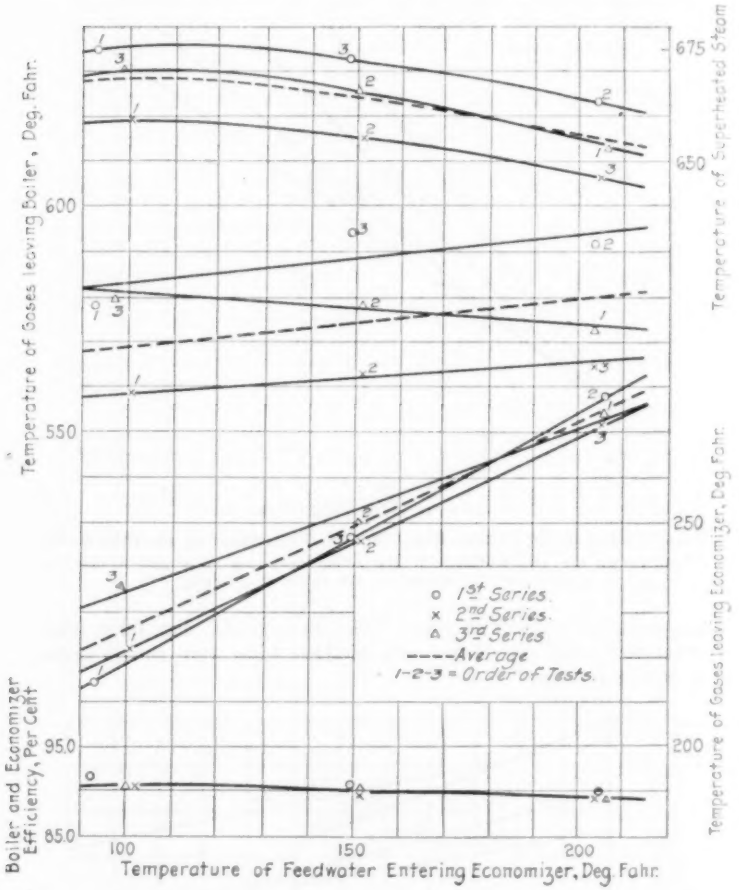


FIG. 9 EFFECT OF CHANGE IN FEEDWATER TEMPERATURE ON BOILER AND ECONOMIZER EFFICIENCY, GAS TEMPERATURE LEAVING BOILER AND ECONOMIZER, AND SUPERHEATED-STEAM TEMPERATURE. TEST DATA CORRECTED TO 12.9 PER CENT CO_2 AND 180 PER CENT OF BOILER RATING

(Average per cent of rating, 180.)

was of two hours' duration with a preparatory period of one hour or longer before each test. The short period of two hours for a boiler test was made possible since all conditions could be held

very steady and since a complete heat balance was not necessary because the only effect of a change in feedwater temperature on the boiler-unit efficiency is due to a change in the economizer-gas leaving temperature, and therefore an accurate determination of the stack loss was all that was required.

Results

11 The results of an increase in steam bleeding are discussed in the following order:

A Effect on turbine-unit performance
B Effect on boiler-unit performance
C Effect on total plant performance as a combination of A and B.

12 A — *Effect on Turbine-Unit Performance.* It will be noted that the changes in turbine-unit performance due to steam bleeding, cited below, agree in general with the conclusions given in previous publications dealing with the subject.

13 The results of an increase in steam bleeding for a constant generator output were

1 An increase in the steam input to the turbine
2 A decrease in the condenser back pressure
3 An increase in the feedwater temperature.

14 The increase in the total water rate naturally follows when the last stages are bypassed by part of the steam, although this effect is partly compensated for by a decrease in the condenser back pressure. Fig. 2 shows the change in water rate with respect to steam bled for different turbine loads.

15 As the amount of steam bled is increased the amount of exhaust steam input to the condenser is decreased, which results in a decrease in the condenser back pressure. Fig. 3 shows the decrease in condenser back pressure with an increase in the amount of steam bled. Fig. 4 shows the effect of the amount of steam condensed on the condenser back pressure.

16 The effect of steam bleeding on the moisture content of the steam in the last stage of the machine and on the turbine efficiency involves a complete discussion of turbine losses, but an analysis of the test results shows that the moisture content of the steam in the last stage is affected more by a change in the condenser back pressure than by a change in the amount of steam bled. The condenser back pressure decreased as the amount of steam bled increased, which resulted in an increase in the moisture content of the steam in the last stage.

17 Fig. 5 shows the change in absolute pressure at the outlet of the 13th stage with different turbine loads and amounts of steam bled.

18 Fig. 6 shows the heat transfer in the bleeder heater as a function of the steam bled.

19 Fig. 7 shows the increase in feedwater temperature with an increase in the amount of steam bled. This is based on the actual operating cycle and includes the exhaust steam from the steam auxiliaries.

20 Fig. 8 shows the change in B.t.u. per kw-hr. chargeable to the turbine unit alone as a function of the steam bled.

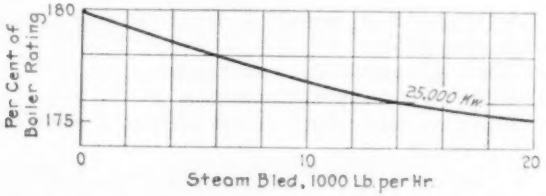


FIG. 10 EFFECT OF STEAM BLEEDING ON PER CENT OF BOILER RATING

21 B — *Effect on the Boiler-Unit Performance.* The effects of an increase in steam bleeding on the boiler unit performance for a constant generator output were:

1 A decrease in the boiler-unit efficiency
2 A decrease in the rate of driving the boiler
3 A decrease in the steam temperature.

22 The decrease in boiler-unit efficiency with an increase in steam bled results from an increase in the feedwater temperature

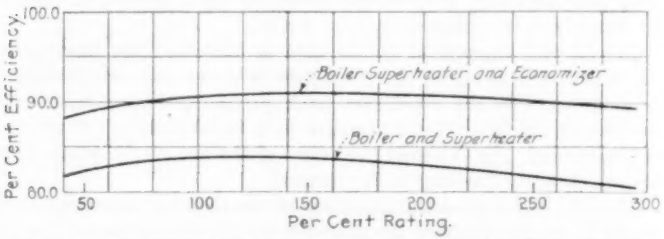


FIG. 11 EFFICIENCY-CAPACITY TESTS OF PULVERIZED-FUEL BOILER

which causes an increase in the economizer exit-gas temperature. Under normal operating conditions, this heat loss in the economizer will be slightly compensated for, due to the fact that the rate of driving at which it is necessary to operate the boiler for a given generator output will decrease as the temperature of the feedwater is increased. Fig. 9 shows the change in boiler and economizer flue-gas temperature, steam temperatures, and unit efficiency with a change in the feedwater temperature, and Fig. 10 shows the change in the rate of driving the boiler for a given generator output with a change in the amount of steam bled.

23 The decrease in steam temperature as the amount of steam bled is increased is caused by both the decrease in the rate of driving the boiler and the increase in feedwater temperature. The decrease of steam temperature with a decrease in the rate of

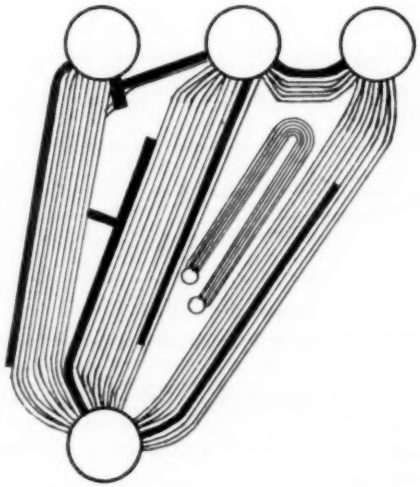


FIG. 12 ARRANGEMENT OF BOILER TUBES

driving the boiler is a well-known characteristic of this type of superheater. The decrease in steam temperature with an increase in feedwater (for constant conditions of rate of driving and excess air) can be visualized by reference to Fig. 12.

24 For a constant rate of driving or horsepower developed by the boiler, the temperature inside of the tubes in the first bank is probably the saturation temperature, irrespective of the feed-

TABLE 1 EFFECT OF STEAM BLEEDING ON TURBINE WATER RATE, FEEDWATER TEMPERATURE, SUPERHEATED-STEAM TEMPERATURE, BOILER RATE, BOILER AND ECONOMIZER EFFICIENCY, AND B.T.U. PER KW-HR. GROSS. LOAD 25,000 KW.

		Water rates, lb. per kw-hr.				Temperature, deg. fahr.				
Steam bled, per hr., lb.	Steam bled, per cent	Turbine	Aux. steam	Misc. steam	Total	Feedwater	Superheated steam	Boiler rate	Boiler and econ. eff.	B.t.u. per kw-hr.
0	0.00	9.71	0.24	0.34	10.29	113.2	668.7	180	90.5	14,392
7,000	2.69	9.83	0.24	0.34	10.41	143.4	663.4	177.7	90.2	14,225
14,000	5.30	9.98	0.24	0.34	10.56	172.7	657.0	176.0	89.8	14,115
20,000	7.48	10.11	0.24	0.34	10.69	197.1	651.3	175.3	89.5	14,021

water temperature; and with a constant heat input to the furnace the temperature and total heat of the gases leaving these tubes and going to the superheater is approximately constant for all feedwater temperatures. Inasmuch as there will be an increase in

FIG. 13 EFFECT OF STEAM BLEEDING ON GROSS B.T.U. PER KW-HR.

the amount of steam generated, as the temperature of the feedwater increases, the amount of heat absorbed per pound of steam passing through the superheater will decrease, resulting in a lower steam temperature.

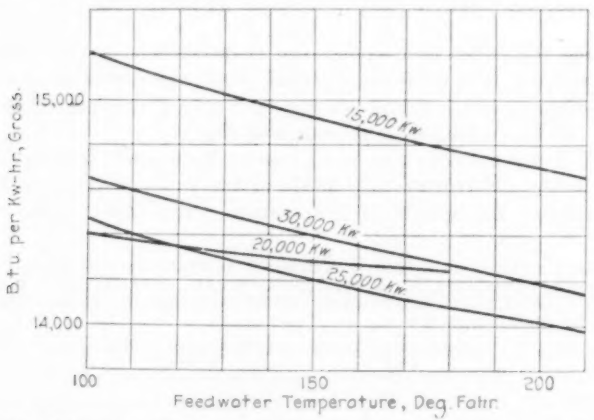


FIG. 14 EFFECT OF CHANGES IN FEEDWATER TEMPERATURE ON GROSS B.T.U. PER KW-HR.

25 C — *Effect on Total Plant Performance.* The B.t.u. per kw-hr. shown in Table 1 is a gross value for a constant generator output and therefore does not include losses incurred in normal varying-load conditions.

26 The last column in Table 1 was calculated from the turbine- and boiler-test results already described, using Figs. 2 and 9, by the method shown in detail in Appendix No. 1. As these figures are representative of the normal operating cycle, the last column

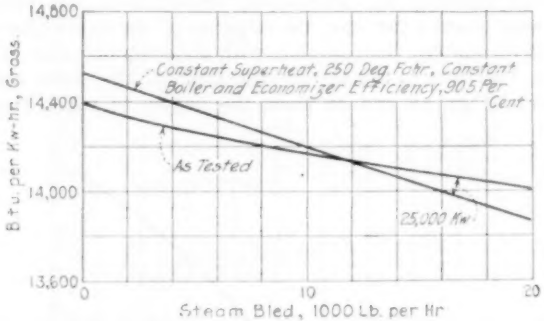


FIG. 15 EFFECTS OF STEAM BLEEDING ON GROSS B.T.U. PER KW-HR. BASED ON CONSTANT SUPERHEAT AND BOILER AND ECONOMIZER EFFICIENCY, COMPARED WITH RESULTS AS TESTED WHICH SHOWED THE CHANGES IN SUPERHEAT AND BOILER-UNIT EFFICIENCY GIVEN IN TABLE 1

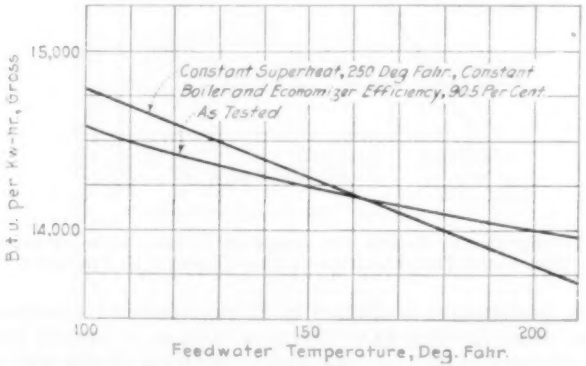


FIG. 16 EFFECTS OF CHANGES IN FEEDWATER TEMPERATURE ON GROSS B.T.U. PER KW-HR. BASED ON CONSTANT SUPERHEAT AND BOILER AND ECONOMIZER EFFICIENCY, COMPARED WITH RESULTS AS TESTED WHICH SHOWED CHANGES IN SUPERHEAT AND BOILER-UNIT EFFICIENCY GIVEN IN TABLE 1

(Load, 25,000 kw.)

is representative of the B.t.u. per kw-hr. gross for steady-load conditions.

27 Fig. 13 shows the change in B.t.u. per kw-hr. gross for different constant generator outputs and amounts of steam bled.

Fig. 14 shows the B.t.u. per kw-hr. gross for different constant generator outputs plotted against feedwater temperatures.

28 It will be noted that the curves for the 20,000-kw. load differ slightly from the others, and it is probable that the four tests run at this load were insufficient to determine the shape of the curve exactly. In determining water rates, the maximum errors were about 1 per cent, the majority of the tests being within one-half of one per cent, so that with a maximum error of 1 per cent for the tests at 20,000 kw. the slope of the curve could be as shown. Figs. 15 and 16 show the degree to which the total

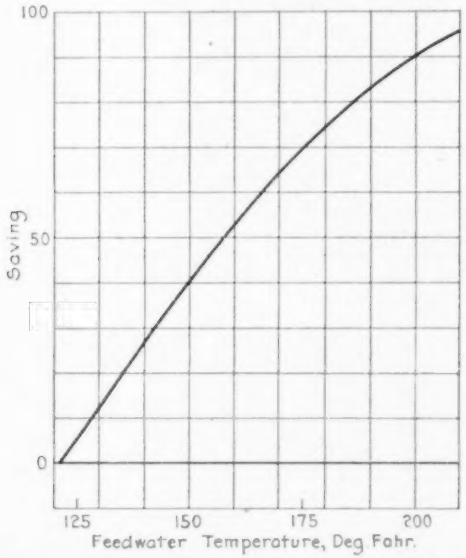


FIG. 17 FINANCIAL SAVING (IN ABSTRACT UNITS) WITH CHANGES IN FEEDWATER TEMPERATURES. FIXED CHARGES, 15 PER CENT

station performance is affected by changes in boiler performance due to steam bleeding, as one curve is based on a constant boiler-unit efficiency of 90.5 per cent and a superheat of 250 deg. fahr. and the other curve is based on actual conditions which resulted in a decrease in boiler-unit efficiency and a decrease in steam superheat due to steam bleeding.

CONCLUSIONS

29 For installations without economizers the change in boiler performance with a change in feedwater temperature will probably be greater than for installations with economizers, as the economizer has the effect of absorbing part of the change in feedwater temperature as shown in Table 2 in which the rise in water

TABLE 2 EFFECT OF CHANGE IN FEEDWATER TEMPERATURE ON BOILER AND ECONOMIZER EFFICIENCY, TEMPERATURE OF GAS LEAVING BOILER AND ECONOMIZER, AND TEMPERATURE OF SUPERHEATED STEAM

VARIABLE FEEDWATER TEMPERATURE TESTS

Test no.	986	1010	1005	988	1006	1009	987	1007	1008
Duration of test, hr	2	2	2	2	2	2	2	2	2
Total water evap., lb. per hr	157,218	164,334	165,579	165,496	169,417	167,569	169,196	177,701	176,270
Per cent of boiler rating	176.0	183.8	185.5	176.9	182.6	180.3	174.1	184.8	183.1
Steam press., lb. per sq. in. gage	241.1	237.8	239.8	242.6	239.3	237.2	243.5	238.0	237.2
Steam temperature, deg. fahr.	670.0	673.2	663.9	667.7	656.7	665.2	656.2	650.1	655.0
Superheat, deg. fahr.	267.0	271.0	261.4	264.2	254.2	263.2	252.2	247.6	253.0
Temp. of feedwater enter. econ., deg. fahr.	93.5	99.0	100.6	150.0	151.6	151.7	204.7	204.9	205.7
Temp. of feedwater leaving econ., deg. fahr.	232.4	235.3	228.0	281.8	267.4	274.2	316.0	302.3	306.2
Temp. rise through econ., deg. fahr.	138.9	136.3	127.4	131.8	115.8	122.5	111.3	97.4	100.5
Temp. of gases leaving boiler, deg. fahr.	573	583	562	590	565	578	585	569	575
Temp. of gases leaving econ., deg. fahr.	211	238	226	242	247	251	272	275	276
Gas temp. drop through econ., deg. fahr.	362	345	336	348	318	327	313	294	299
CO_2 in economizer gases, per cent	12.89	12.87	13.25	12.81	13.25	12.75	12.93	13.09	13.11
O_2 in economizer gases, per cent	6.30	6.40	5.95	6.51	5.95	6.51	6.33	6.09	6.12
					Per cent				
Heat absorbed, boiler and economizer	91.32	90.57	91.02	90.44	90.44	90.17	89.65	89.63	89.61
Loss in dry flue gases	3.68	4.39	3.96	4.51	4.50	4.77	5.25	5.27	5.29
Loss due to hydrogen in coal	3.72	3.76	3.74	3.77	3.78	3.78	3.82	3.82	3.82
Loss due to moisture in coal	0.28	0.28	0.28	0.28	0.28	0.28	0.28	0.28	0.28
Loss due to radiation and unaccounted for	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
Total	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00

temperature through the economizer is seen to decrease as the temperature of the feedwater increases.

30 The effect of a change in superheat due to steam bleeding pertains more to existing installations than to the design of new plants, as in the latter case the superheater can be designed for the feedwater temperature decided upon.

31 The financial saving due to bleeding steam from the 13th stage of the 17-stage machine is shown in Fig. 17. This is calculated

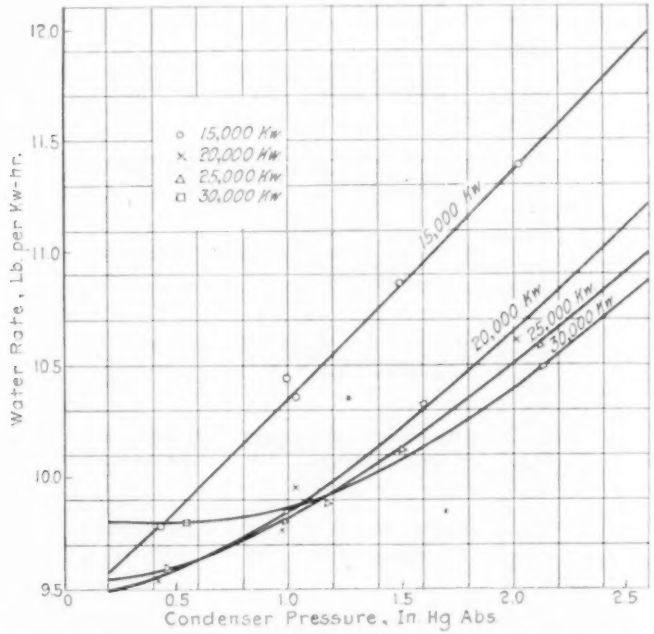


FIG. 18 EFFECT OF CONDENSER BACK PRESSURE ON TURBINE WATER RATE

(Steam pressure, 250 lb. per sq. in. abs.; superheat, 250 deg. fahr.)

for a load of 25,000 kw. and a load factor of 70 per cent. Fifteen per cent of the total cost of installing the bleeder heater is deducted from the gross saving to give the net saving. This curve is not extended beyond 210 deg. fahr. because, for the installation tested, an increase in feed temperature above this amount would involve additional equipment and investment charges, as this installation was designed as an extension of a large plant of older design.

32 Although an extension of this curve is a theoretical extrapolation and naturally an inaccurate procedure, it is of interest to note that for similar equipment which would not involve additional investment charges for higher feedwater temperatures

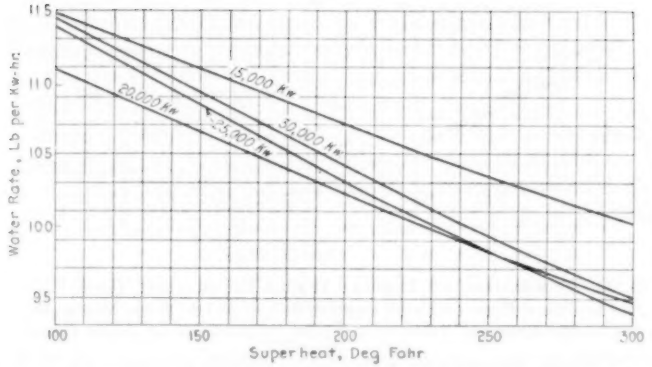


FIG. 19 EFFECT OF CHANGE IN SUPERHEAT ON TURBINE WATER RATE
(Initial steam pressure, 250 lb. per sq. in. abs.; condenser pressure, 1 in. Hg abs.)

the economical feed temperature would probably be below 250 deg. fahr.

33 As the purpose of the paper is to show test results only, extensive calculations on power-plant cycles and the amount of

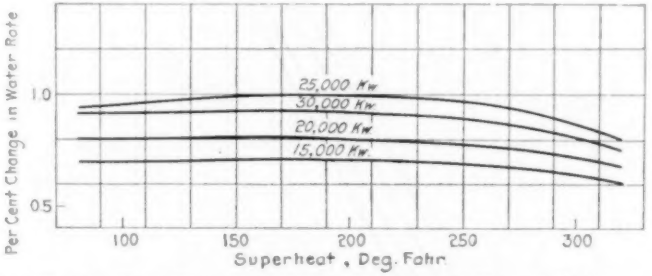


FIG. 20 PER CENT CHANGE IN WATER RATE FOR 10-DEGREE CHANGE IN SUPERHEAT
(Condenser pressure, 1 in. Hg abs.; steam pressure, 250 lb. abs.)

bleeding to use are omitted, but it is hoped that it indicates the value to give the different factors to be considered in determining the economical feedwater temperature.

PART II

TURBINE, CONDENSER, AND GENERATOR CHARACTERISTICS

34 Fig. 18 shows the change in turbine water rate at constant loads, constant pressure, and constant superheat with variable back pressure. It is interesting to note that at the low back pres-

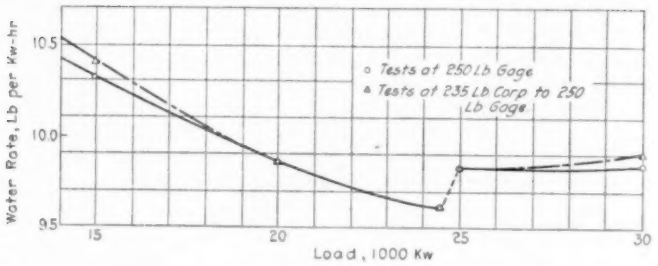


FIG. 21 COMPARISON OF TURBINE WATER RATES WITH TESTS RUN AT 235 LB. GAGE PRESSURE AND CORRECTED TO 250 LB. GAGE, USING MANUFACTURERS SPECIFICATIONS AND TESTS RUN AT 250 LB. GAGE PRESSURE

(Superheat, 250 deg.; pressure, gage, 250 lb.; condenser pressure, 1 in. Hg abs.)

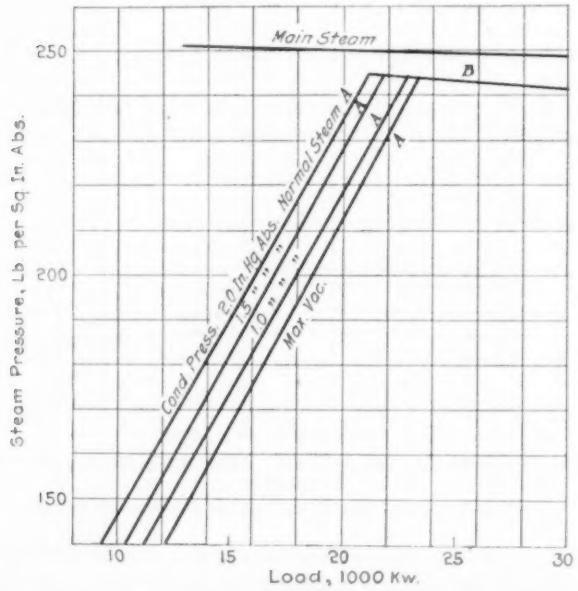


FIG. 22 EFFECT OF CONDENSER BACK PRESSURE ON TURBINE HEAD PRESSURE

sures of 0.4 in. of mercury there is still a decrease in water rate with a decrease in back pressure for loads up to 25,000 kw., but that the limiting vacuum for a load of 30,000 kw. is about 0.7 in. of mercury.

35 Figs. 19 and 20 show the change in turbine water rate due to a change in the degree of superheat. It is of interest to note

that the commonly used figure of 1 per cent correction for ten degrees change in superheat was also shown by the tests. In drawing these curves the actual data were corrected to constant conditions of back pressure and generator power factor.

36 The majority of the tests were run with a steam pressure of 235 lb. per sq. in. gage, but one series was run at 250 lb. per sq. in. gage. The tests indicate that the pressure-correction data furnished by the manufacturer of 0.2 of 1 per cent for a change

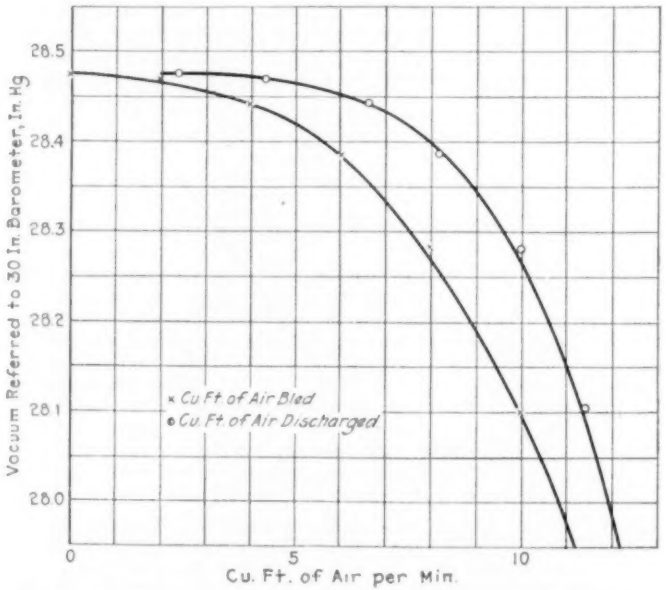


FIG. 23 EFFECT OF AIR IN CONDENSER ON CONDENSER VACUUM

(The air bled was measured as it entered the turbine exhaust casing. The air discharged was measured as discharged from the steam-jet air ejector. Load, 25,000 kw.)

of 10 lb. per sq. in. before the load at which the overload valve opens, and 0.4 of 1 per cent for a change of 10 lb. per sq. in. above the load at which this valve opens is correct. This is shown in Fig. 21 by the fact that the tests at 250 lb. per sq. in. gage coincide with the tests at 235 lb. per sq. in. gage corrected to 250 lb. gage.

37 The step in the water-rate curve occurs at the point at which the overload valve opens. This point is influenced by both the initial steam pressure and the condenser back pressure. Referring to Fig. 22, the overload valve opens approximately where the lines marked *A* meet the line marked *B*. The lines marked *A*

represent the turbine head pressure or the pressure between the throttling valve and the turbine. The line marked *B* represents the head pressure when the throttling valve is wide open. As shown by the curves a decrease in the condenser back pressure causes the step in the curve to occur at a higher load. An increase in the main steam pressure has the effect of raising the line *B*

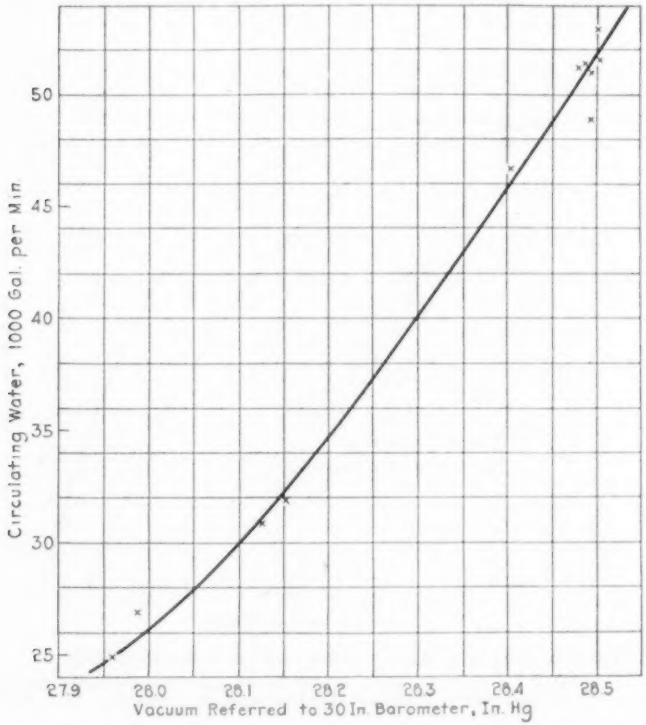


FIG. 24 EFFECT OF QUANTITY OF CIRCULATING WATER PUMPED ON CONDENSER VACUUM

vertically and causing the step in the curve to occur at a higher load.

38 Fig. 23 shows the change in condenser back pressure due to a change in the air content of the turbine exhaust steam. Air was bled into the turbine exhaust casing through nozzles which had been calibrated by an orifice. The air removed from the condenser by steam-jet ejectors was measured by an orifice placed in the discharge from the ejector.

39 Fig. 24 shows the change in condenser back pressure due to a change in the amount of circulating water. The amount of circulating water was calculated from a heat balance on the condenser.

APPENDIX NO. 1

CALCULATION OF TABLE 1

40 From Fig. 2 the turbine water rates at 25,000 kw. generator output, 250 deg. fahr. superheat, and various amounts of steam bleeding as indicated were obtained. The steam auxiliary water rate and the miscellaneous steam water rate were average values for this installation. The sum of the turbine water rate, the auxiliary water rate, and the miscellaneous steam water rate equals the total water rate. The feedwater temperatures were calculated by the heat-balance method as follows:

$$(T-32) = \frac{T_E h_C + W_A H_A + W_M h_M}{W_E} \text{ for zero bleeding}$$

and

$$(T-32) = \frac{T_E h_C + W_A H_A + W_M h_M + \frac{W_B h_B}{25000}}{W_E} \text{ for } W_B \text{ pounds of steam bled,}$$

where T = temperature of the feedwater
T_E = turbine water rate, lb. per kw-hr.
W_A = auxiliary water rate, lb. per kw-hr.
W_M = miscellaneous steam water rate, lb. per kw-hr.
W_E = total water rate, lb. per kw-hr.
W_B = steam bled, lb. per hr.
h_C = heat in condensate leaving generator air cooler, B.t.u. per lb.
H_A = heat in auxiliary exhaust, B.t.u. per lb.
h_M = heat in miscellaneous water, B.t.u. per lb.
H_B = heat in bled steam, B.t.u. per lb.

Using the feedwater temperatures calculated in this manner in conjunction with Fig. 9, the temperatures of the superheated steam at 180 per cent of boiler rating were obtained. The turbine water rates obtained from Fig. 2 were then corrected for these steam temperatures, using the correction curve as shown in Fig. 19. Then by the heat-balance method as before, the feedwater temperatures were recalculated, using the corrected water rates.

41 Knowing the percentage of boiler rating with zero bleeding to be 180, the percentages of boiler rating for the various amounts of steam bleeding were calculated as follows:

$$R = Wh/C$$

where R = per cent of boiler rating
W = total water rate, lb. per kw-hr.
h = heat added by boiler and superheater, B.t.u. per lb.
C = constant.

42 The turbine water rates having been corrected for variations in steam temperature due to change in feedwater temperature, were then corrected for variations in steam temperature due to changes in per cent of boiler rating and the feedwater temperatures recalculated as before.

43 Using these final corrected values for feedwater temperature, the boiler and economizer efficiencies were obtained from Fig. 9. The B.t.u. per kw-hr. was then calculated as follows:

$$\text{B.t.u. per kw-hr.} = Wh/E$$

where W = total water rate in lb. per kw-hr.
h = heat added by boiler, superheater, and economizer, B.t.u. per lb.
E = boiler and economizer efficiency expressed as a decimal.

APPENDIX NO. 2

LOGS OF TESTS

44 Tables 3, 4, and 5, giving logs of the tests conducted, will be found on the two pages next following.

DISCUSSION

M. K. Drewry.[1] Evaluating and weighting all the benefits and detriments of regenerative feedwater heating is no little task. The author is to be commended upon the completeness and thoroughness of the work which he reports. There is much valuable information in his paper.

Whether extra moisture is removed with extraction steam is a question which it was hoped might be answered in the paper. Study of the data and a knowledge of the type of turbine tested point out that all steam extracted was superheated at the time it left the turbine, explaining why no conclusions on the above subject were reached.

That extraction steam taken from the 14th stage of the same type of unit as described in this paper contains no more moisture than steam passing the extraction point, was learned by a test on a 17-stage, 30,000-kw. unit at the Lakeside Station. Expansion lines of the unit indicated that steam with 2 per cent moisture existed at the stage, yet none of the equivalent 5000 lb. per hr. of condensate entrained in the steam flow passing the extraction point was removed with the extraction steam. No check test has been performed, yet in the original test all quantities were measured with apparent accuracy. The experience of others in trying to detect this condition claimed to increase turbine efficiency by moisture elimination would be interesting.

Heat transfers of 750 B.t.u. per deg. fahr. per sq. ft. per hour are obtained daily in the several single-extraction heaters installed during the summer of 1925 at the Lakeside plant. The maximum values reported in the paper are 60 per cent of 750. It is gathered from analysis of the data that the amount of steam bled was controlled by valve throttling, and not by variation in venting the

[1] Assistant Chief Engineer of Power Plants, Lakeside Power Plant, Milwaukee Electric Railway & Light Co., Milwaukee, Wis. Jun. A.S.M.E.

TABLE 3 PARTIAL LOG OF BLEEDER HEATER TESTS

Test number	949	948	985	947	950	955A	956	938	954	936	984	953	952	951	946	945A	944	943	942
Kilowatts generated	14929	14875	15032	14982	20117	19983	19749	19903	24959	24831	24911	24918	24713	24958	29854	29865	29828	29946	29759
Actual water rate, lb. per kw-hr.	10.042	10.174	10.226	10.231	9.694	9.723	9.795	9.925	9.677	9.818	9.890	9.866	9.916	9.971	9.999	9.924	10.059	10.099	10.336
Vacuum refd. to 30 in.	29.215	29.263	29.214	29.268	29.242	29.240	29.244	29.208	29.124	29.157	29.176	29.154	29.164	29.180	29.004	29.007	29.019	29.023	29.036
Steam, press. main, lb. per sq. in. abs.	248.9	251.3	248.8	250.4	248.3	250.5	249.3	246.8	249.0	246.6	252.7	249.6	248.3	249.0	247.6	248.4	247.4	247.6	244.9
Superheat, deg. fahr.	263.8	261.8	270.6	262.5	256.0	262.1	271.1	265.0	271.7	268.9	256.5	262.9	268.7	265.4	259.0	273.1	266.6	264.1	252.3
Normal press., lb. per sq. in. abs.	167.6	170.3	176.2	172.4	216.8	218.2	218.6	223.0	241.4	240.0	245.2	241.4	240.9	240.8	238.5	239.4	237.8	238.3	235.3
Bled Steam																			
Press. leaving turb., in. Hg abs.	.....	15.16	16.42	14.32	.....	20.18	19.5	18.7	.....	24.91	24.33	24.44	23.68	23.21	.....	31.25	30.75	29.77	28.73
Temp. leaving turb., deg. fahr.	.....	180.6	178.5	177.7	.....	193.3	191.4	188.9	.....	203.5	202.4	202.4	201.8	199.7	.....	223.7	217.8	213.1	209.5
Press. entering heater, in. Hg abs.	.....	7.8	11.88	12.58	.....	6.86	8.8	14.08	.....	10.97	11.73	12.02	13.34	15.96	.....	8.31	12.70	14.85	18.07
Temp. entering heater, deg. fahr.	.....	155.8	169.3	166.8	.....	152.2	159.4	177.7	.....	.....	171.4	168.2	172.9	182.5	.....	203.8	193.4	178.9	187.4
Temp. cond., deg. fahr.	.....	121.3	149.1	149.6	.....	114.2	125.2	146.5	.....	125.0	136.1	137.0	142.6	156.6	.....	118.5	126.1	143.4	160.6
Condensate																			
Temp. entering heater, deg. fahr.	89.1	88.0	89.4	88.8	86.2	85.4	85.7	85.0	86.9	85.2	86.5	85.8	85.8	85.2	88.3	88.0	87.1	86.8	86.2
Temp. leaving heater, deg. fahr.	.....	119.9	148.3	151.6	.....	106.3	124.1	144.3	.....	121.0	136.8	136.7	136.0	156.5	.....	108.5	124.1	145.1	163.3
Temp. rise through heater, deg. fahr.	0	31.9	58.9	62.8	0	20.9	38.4	59.3	0	35.8	50.3	50.9	60.2	71.3	0	20.5	37.0	58.3	77.1
Initial steam, B.t.u. per lb.	1345	1345	1348	1344	1340.5	1344	1348	1345	1349	1346	1342	1344	1347	1345.5	1342	1349	1346	1345	1338.5
Bled steam, B.t.u. per lb.	.....	1128	1134	1125	.....	1126	1129	1126	.....	1137	1135	1134	1135	1132	.....	1150	1145	1138	1138
Exhaust steam, B.t.u. per lb.	976	976	973.2	971.8	965	966.3	969.7	969.1	975	977.3	969	971.5	973	972.3	984	985.3	981.9	982.6	980.2
Power factor, per cent	80.6	80.8	64.5	75.0	76.6	79.0	75.3	82.7	78.0	86.3	81.5	82.6	79.3	81.7	88.8	90.3	87.2	94.0	90.0
Generator eff., per cent	95.2	95.2	94.5	95.1	96.1	96.2	96.1	96.2	96.4	97.8	96.6	96.7	96.5	96.6	97.0	97.1	96.9	97.3	97.7
Input to generator, kw	15690	15633	15915	15757	20929	20783	20561	20685	25891	25403	25788	25776	25617	25836	30777	30763	30782	30793	30657
Input to exciter, kw	53	53	73	59	56	69	74	65	87	73	83	78	86	82	82	67	87	69	79
Input to bearings and radiation, kw.	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375
Input to fans, kw	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95
Total turbine output, kw	16213	16156	16458	16286	21455	21322	21105	21220	26448	25946	26341	26324	26173	26388	31329	31300	31339	31332	31206
Bled steam, lb. per hr.	0	4648	8903	9556	0	3890	7171	11580	0	8360	12020	12194	14401	17613	0	5713	10509	17175	23494
Quality of exhaust steam, per cent.	89.1	89.1	88.8	88.8	88.0	88.1	88.5	88.4	88.8	89.0	88.3	88.5	88.6	88.6	89.5	89.5	89.3	89.4	89.1

TABLE 4 LOG OF VARIABLE VACUUM TESTS

Test number	824	827	837	838	840	823	826	841	835	836	822	825	829	830	831	821	828	833	834
Kw-hr. generated	14623	14879	14855	14874	14855	20146	19953	20091	19899	19897	24583	24865	24888	24859	24971	30103	29908	29580	29865
Actual water rate, lb. per kw-hr..	9.776	10.433	10.411	10.810	11.424	9.554	9.781	9.790	10.259	10.513	9.692	9.827	9.937	10.106	10.526	9.875	9.869	10.207	10.439
Vacuum refd. to 30 in.	29.564	29.017	28.971	28.506	27.975	29.580	29.039	29.010	28.480	27.988	29.540	29.017	28 833	28.501	27.897	29.452	29.007	28.407	27.868
Steam press. main, lb. per sq. in. abs.	246.2	249.4	250.3	251.2	254.1	246.5	248.0	251.0	251.2	250.5	246.6	245.2	249.4	247.9	248.6	246.0	249.1	249.4	247.1
Superheat, deg. fahr.	255.0	254.1	248.4	259.0	248.3	249.5	250.2	250.8	254.9	263.7	239.4	250.0	243.5	251.8	257.7	246.3	249.3	263.6	256.2
Normal press., lb. per sq. in. abs..	162.5	172.6	173.3	181.2	188.6	213.5	216.1	217.9	226.5	233.3	241.5	240.4	243.8	242.5	243.4	239.4	242.0	242.1	239.9
Initial steam, B.t.u. per lb.	1341	1340.5	1338	1344	1337.8	1338	1339.0	1339.5	1341.0	1345.0	1333.0	1338.0	1336.0	1340.2	1341.5	1336.5	1338.0	1345.0	1341.0
Exhaust steam, B.t.u. per lb.	961.2	981.7	981.3	999.7	1012.2	955.8	966.4	968.0	985.0	998.7	960.8	971.5	973.6	984.1	998.5	975.4	976.4	994.2	999.4
Power factor, per cent.	76.0	63.0	73.4	71.0	70.6	72.3	77.5	84.0	73.6	83.0	82.3	85.3	87.3	90.0	80.3	96.0	95.0	89.3	99.0
Generator eff., per cent.	95.1	94.3	95.0	94.8	94.9	95.9	96.1	96.2	96.9	96.2	96.7	96.8	96.8	96.8	96.6	97.4	97.3	97.0	97.4
Input to generator, kw.	15380	15778	15637	15690	15662	21007	20763	20881	20652	20689	25435	25700	25711	25681	25863	30922	30738	30482	30678
Input to exciter, kw.	59	70	57	59	59	77	71	60	76	62	77	71	67	62	80	60	63	80	51
Input to bearings and radiation, kw.	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375
Input to fans, kw.	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95
Total turbine output, kw.	15909	16318	16164	16219	16191	21544	21304	21411	21198	21221	25982	26241	26248	26213	26413	31452	31271	31032	31199

TABLE 5 LOG OF VARIABLE SUPERHEAT TESTS

Test number	961	960	968	972	962	959	967	971	963	958	966	970	964	957	965	969
Kw-hr. generated	15269	14948	15005	14988	20165	20007	19936	20202	25128	24555	24764	24778	29608	29971	29877	29854
Actual water rate, lb. per kw-hr..	11.150	10.717	10.453	10.140	10.836	10.296	9.993	9.565	10.973	10.439	9.981	9.512	11.162	10.491	10.188	9.549
Vacuum refd. to 30 in.	29.013	29.056	29.033	29.064	29.077	28.994	28.982	29.048	29.044	29.083	29.000	29.086	28.904	28.976	28.868	28.979
Steam press., main, lb. per sq. in. abs.	246.3	248.7	250.5	256.5	246.7	248.9	250.0	256.3	246.1	250.2	248.7	253.0	246.0	247.2	250.6	253.0
Superheat, deg fahr	120.8	193.6	237.0	271.7	123.2	197.3	245.6	272.4	126.4	193.4	245.1	277.5	124.3	193.5	243.2	278.7
Normal press., lb. per sq. in. abs..	185.4	175.5	178.1	171.5	228.2	224.3	221.5	219.2	239.4	242.8	240.6	245.2	236.9	238.4	241.1	243.5
Initial steam, B.t.u. per lb.	1274	1310.5	1332	1349	1275	1312.5	1336.5	1349.5	1276	1311	1336.5	1350.5	1275	1310.5	1336	1353
Exhaust steam, B.t.u. per lb.	942.2	964.9	974.4	982.5	939.6	959	970.3	969.4	948.5	965.7	975.4	972.5	954.8	969.5	983.8	979.2
Power factor, per cent.	81.8	86.2	64.2	70.8	88.2	81.0	70.7	82.3	95.3	86.0	85.3	94.0	92.7	90.0	85.7	95.7
Generator eff., per cent.	95.3	95.2	94.5	94.9	96.3	96.2	95.8	96.2	96.9	96.7	96.7	96.9	97.2	97.1	96.9	97.3
Input to generator, kw.	16030	15702	15887	15793	20946	20797	20821	20993	25932	25393	25604	25584	30461	30866	30849	30682
Input to exciter, kw.	53	48	77	57	57	67	84	57	54	71	77	51	72	79	93	74
Input to bearings and radiation, kw.	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375	375
Input to fans, kw.	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95
Total turbine output, kw.	16553	16220	16434	16320	21473	21334	21375	21520	26456	25934	26151	26105	31003	31415	31412	31226

heater, which latter would explain the relatively poor heat transfers. The low condensate temperatures from the hotwell of the heater (representing a minimum of 17 deg. refrigeration in the last test reported, No. 942) indicate improper or insufficient air removal from the heater. Terminal temperature differences of 5 deg., and refrigerations of 2 deg., have been observed repeatedly with test instruments at the Lakeside plant.

The third item of Table 3 shows a water-rate increase (above non-extracting results of test No. 946) of 3.36 per cent when feedwater is heated to 163.3 deg. fahr. Were the same heating done with extraction from the next lower stage (the 14th), as might be done could the 10.7 lb. drop from turbine to heater be reduced and the heater made to perform better, then a water rate nearly $\frac{1}{2}$ per cent less could be obtained. The importance of eliminating excessive pressure and temperature drops is thus illustrated. Use of several extraction heaters to obtain a given final temperature, besides being theoretically more efficient, is also additionally efficient because lower pressure drops may be obtained without resorting to excessive pipe sizes.

The manufacturers' pressure-correction rate of 0.4 per cent per 10 lb. mentioned in Par. 36 is considerably less than theoretical gains. Actual test results at 30,000-kw. loads at both Cleveland and Milwaukee show the actual gain about coincident with the theoretical gain, or $\frac{3}{4}$ per cent per 10 lb., at loads above the most economical rating.

C. G. Spencer.[1] During the past three years much has been written on heat balances and bleeder heaters with widely different arrangements and predicted performances. The author presents a compilation of test results showing not only the effect of single-stage bleeding on a 30,000-kw. unit, but also the influence of this bleeding on boiler and condenser performance. This is a valuable contribution and the first that has come to the writer's notice in which this has been done.

It is quite generally accepted that economizers have no place in a design where the feedwater is heated by steam bled from the main unit. In this case, however, with a 72 per cent economizer and one-stage bleeding, there is shown an overall station improvement due to bleeding of 371 B.t.u. per kw-hr., or 2.6 per cent. The feedwater temperature is raised from 115 deg. to 215 deg. fahr. in the bleeder heater as shown in Fig. 7. While this results in a lowering of boiler and economizer efficiency of about one point, the improvement in turbine efficiency overcomes this and results in the station improvement given.

Heat-balance calculations for bleeder heaters generally disregard the effect of bleeding on the condenser performance. Fig. 3

[1] Engineer, McClellan & Junkersfeld, Inc., New York, N. Y. Mem. A.S.M.E.

shows an improvement in vacuum of about 0.4 in. at full load and 0.5 in. at 25,000 kw. These are interesting facts which may be applied generally. The effect on the boiler and economizer is also interesting, but difficult to predict at the time heat-balance calculations for a design are made.

F. M. Van Deventer.[1] The flow diagram which the author showed in illustrating his paper indicated that the low-pressure heater was drained through a loop seal into the hotwell of the main condenser. It would be pertinent to know whether any difficulty has been experienced in the operation of this drain system.

The motive for this discussion is an unsatisfactory experience with an installation of three stages of regenerative feedwater heating in which the low-pressure heater was drained through a loop seal into the condenser hotwell. When this installation was put into operation, it was found that the seal was not functioning properly and that steam was passing continuously from the heater into the main condenser. A careful check-up showed that the depth of the seal was about 25 per cent greater than the head of water required to balance the difference in pressure between the heater shell and condenser.

To determine just what happens in this type of seal a thermodynamic study is pertinent. The condensate from the extraction heater has a temperature a few degrees below that of saturated steam at the pressure existing in the heater shell. The total heat of each pound of water drained from the heater, therefore, is equal to the "heat of the liquid" corresponding to the temperature of the drain. As the condensate emerges from the loop seal it is then under the pressure existing in the main condenser (ignoring pipe friction), and each pound can contain only the "heat of the liquid" corresponding to the saturation temperature for condenser pressure. Since this quantity is considerably less than the heat of the liquid under heater conditions, the difference of these two quantities represents the quantity of heat which is released from the liquid and must be absorbed by the evaporation of a certain portion of the liquid into vapor. This "flashing" is exactly equivalent to that which takes place in the blow-off from a boiler, in which case approximately 20 per cent of the water blown flashes into vapor. In the case of the loop seal, the water rises from the bottom of the loop to the top, passing progressively from a region of high pressure to a region of low pressure, and vapor will be released throughout the entire length of the discharge leg of the seal. Thus the seal will be filled not with water alone, but with a mixture of water and vapor, the density of which must be materially less than the density of an equivalent column of water. The

[1] Mechanical Engineer, Construction Department, Henry L. Doherty & Co., New York, N. Y. Mem. A.S.M.E.

theoretical amount of vapor formed in this flashing process is quite small when considered by weight, namely, from 5 to 12 per cent, depending upon the pressure in the heater and vacuum in the condenser. However, when the high specific volume of steam at one and two inches absolute pressure is taken into account (330-660 cu. ft. per lb.) it is found that the theoretical volume of steam formed is from 1000 to 6000 times the volume of water entering the seal and it is obvious that the density of the mixture in the seal will be many times less than that of water, and that the action will be similar to that of an " air lift."

The foregoing analysis convincingly explains the failure of the loop seal referred to, and it has since been learned that a number of other companies have experienced a similar difficulty.

There are several methods of alleviating this condition. Probably the most common method of draining the low-pressure heaters in present-day use is that of using a drain pump which discharges the condensate into the feedwater system at a point of higher pressure instead of a point of lower pressure. So long as the pressure is not reduced, there will not be any tendency to flash. This method has a thermal advantage in that all the heat in the condensate enters the feedwater cycle, whereas in the case of the loop seal or any drain system whereby the condensate from the heater is discharged to the main condenser shell or hot well, the heat in the vapor resulting from the flashing is absorbed by the circulating water of the main condenser and is lost from the feedwater heating system. The disadvantage of the pump-drainage system is the fact that the pump is a rotating or reciprocating unit, and many operators are desirous of limiting the number of machines with moving parts to a minimum.

Another remedy is to increase the depth of the loop seal so as to counteract the decrease in density of the column of water and vapor mixture in the seal. The difficulty with this, however, is that it cannot be predetermined just how deep such a seal must be. In the light of the foregoing thermodynamic analysis, it is seen that theoretically a seal several hundred times the normal depth might be required. Increased depth, however, has been successful in some instances when accompanied by the introduction of cold water into the seal. This cold water depresses the entering temperature so that flashing, while not entirely eliminated, is reduced.

A third remedy is the addition of a heat interchanger in the condensate line from the heater. By passing the feedwater through this heat interchanger before it enters the stage heater, it is possible to reduce the temperature of the condensate to within a few degrees of the main condensate temperature, in which event the thermal head through which flashing occurs is reduced to a minimum. The heat interchanger also has the advantage enumerated for the pump, namely, that the heat which would be lost due to flashing is absorbed by the feedwater and not lost through the

main unit circulating system. An economic study, however, indicates that the fixed charges on a heat interchanger of the necessary size would approximately equal the value of the heat reclaimed. Consequently the use of such apparatus is hardly justified in most cases.

The most logical remedy would be to design the heater so as to incorporate an amount of surface equivalent to the heat interchanger in the stage heater itself. Reference may be made here to the development of the modern condenser. Early condensers were so built that most of the work was done by a relatively small portion of the surface in the condenser. The condensate then trickled over the remaining surface and its temperature was often depressed many degrees below that corresponding to the vacuum. This, of course, was a thermal and economic loss. Modern condenser design provides steam lanes into the body of the condenser so that practically the entire surface is effective as condenser surface and the refrigeration of condensate is reduced to a minimum. Heater design, in the light of condenser design, should consider the manner in which the condensate is to be handled. If the low-pressure heater is to be drained by a pump and the condensate discharged into a point of higher pressure, then the heater should be designed as a modern condenser and the temperature of the condensate made as high as possible. If, however, the heater is to be drained through a loop seal, then it should be designed as an old-fashioned condenser and the distribution arranged so as to obtain maximum refrigeration or temperature depression of the condensate.

We have recently revised the design of a heater on the principle just described, and although no data are available at the present time, we believe that satisfactory operation will be obtained therefrom.

The foregoing applies primarily to the low-pressure heaters only. In the case of extraction heaters operating on higher pressure, they would be drained either by pumping to a higher pressure or through a loop seal into a heater of lower pressure. In such cases the specific volume of steam at the lower pressure is not sufficiently high to cause a material "air lift" action in the seal. It is advisable, however, in all cases to provide more depth in the loop seal than is theoretically required when considered on a water basis.

W. J. WOHLENBERG.[1] The writer does not believe that the author's statement in Par. 29 is quite correct. According to this statement, there would be more effect on the performance of the boiler with a change in feedwater temperature without an economizer than would be the case with an economizer. If that were

[1] Associate Professor of Mechanical Engineering, Yale University, New Haven, Conn. Mem. A.S.M.E.

true there never would be any point in adding economizer surface as such.

Concerning the influence of feedwater temperature when comparing its effect on the performance of a boiler as against performance of a boiler with economizer, the following appear to be the conditions. Feedwater temperature influences the escaping gas loss indirectly through its effect on boiler or economizer surface temperatures with which it comes in contact. If the feedwater enters an economizer (which is merely part of the feedwater system) the mean surface temperature will be raised or lowered as the entering feedwater temperature is raised or lowered. The feedwater circuit through the economizer is not in the main boiler circulation path and consequently the water contained in the economizer is not influenced by the temperature of the water in the boiler proper.

In the other case, when no economizer is used, the feedwater is discharged directly into the water-circulating system within the boiler. Several years ago the writer investigated the circulation conditions in a Bigelow boiler at the point at which feedwater enters, and discovered that the rate of circulation is many times the rate of feedwater supply, even when feedwater is supplied at a high rate. The result of this action is an almost immediate distribution of the incoming feedwater with such relatively large quantities of water already in the boiler that the temperature of the mixture is practically at the boiling point corresponding to the pressure in the boiler. Under these conditions the boiler surface temperatures are hardly influenced at all by the temperature changes in the incoming feedwater. Since the gases give off heat directly to the surface, the escaping gas loss obviously can hardly be influenced under such conditions by a change in feedwater temperature. If this is true the boiler efficiency is not influenced appreciably by changes in boiler-feedwater temperature. Apparently the lower incoming feedwater temperature will reduce the rate of steam formation in the rear sections of the boiler to some extent, but without influencing the temperature of the boiler surface appreciably.

THE AUTHOR. The statement that all the steam extracted from the turbine was superheated at the time it left the turbine is not borne out by the test results, which show that none of the bled steam for these tests was superheated and that the quality ranged from 98.2 to 99.0 per cent. A study of the expansion characteristics of this turbine will also show that none of the bled steam was superheated.

The heat-transfer rate of the bleeder heater in question cannot be increased for given condensate temperatures and turbine load and steam conditions by any change in the operation of the heater, with the exception of passing a large amount of steam through

the vent to the main condenser, which would be bad practice as the heat in this steam would be lost to the circulating water. For the first condition, with no steam going through the vent, let the transfer rate $= \frac{W(H-h)}{ST}$,

where W = weight of bled steam
H = heat content per pound of bled steam
h = heat content per pound of condensed bled steam
S = heating surface
T = mean temperature difference.

The final feedwater temperature desired determines the amount W of steam bled. Each pound of this bled steam has a heating value H determined by the turbine characteristics, and is not changed by the throttling in the line to the heater. The change in h from no refrigeration to the value obtained is only 27, and is not sufficient to materially change the heat-transfer rate. The quality of the bled steam for one point (test 942) was 98.7 per cent leaving the turbine and 99.7 per cent entering the heater. The temperature in the first case was 209 deg. fahr. and in the second case, 187 deg. fahr. Had there been no throttling action from the turbine to the bleeder heater, the value T would have been increased and the transfer rate decreased, since the initial and final condensate temperatures remain unchanged for a given amount of steam bled.

Since the values W and $(H-h)$ are not determined by the bleeder heater or its operation, the rate of heat transfer for a given amount of steam bled cannot be increased by any possible change in the operation of the heater. The curve in Fig. 6 shows that high heat-transfer rates can be obtained by bleeding larger amounts of steam. This was not done, as the steam bled was sufficient to obtain the feed temperatures desired.

Higher heat-transfer rates can also be obtained by using a smaller heater. The present heater was chosen sufficiently large to take care of possible future changes in auxiliaries. The transfer rate, of course, cannot be indefinitely increased by decreasing the size of the heater S, but for each heater there is a maximum transfer rate for given steam and water-inlet temperatures which cannot be exceeded without passing some of the steam through the heater without being condensed. This maximum transfer rate is limited by the character of the heater design and the amount of non-condensable vapors handled.

The author does not intend to imply that throttling action need not be considered in designing bleeder-heater installations. It is an important factor in determining the proper stage from which to bleed steam, but as this has been very well brought out in the previous papers on the theory and design of steam-bleeding installations, it does not require repetition in this paper.

As the heat in any steam which passes from the bleeder heater by means of a vent to the main condenser is lost to the circulating water, these vents should be of minimum size. The heaters at the Lake Shore Station give identical results whether they are vented or not, which is an indication that the air leakage is small.

Where condensed bled steam is returned to the condenser hotwell, the temperature of this condensate should be as low as possible, as it flashes into steam before entering the condenser and gives up a small amount of heat to the circulating water. For the heater in question a maximum refrigeration is therefore desirable.

That boiler efficiency is not materially affected by a change in the boiler feedwater temperature, whereas the combined efficiency of a boiler and economizer unit is materially affected by a change in the economizer feedwater temperature, is borne out by the test results. The point the author had in mind in this connection was that for a given change in feedwater temperature to the unit, there is a greater change in steam temperature for a boiler alone than for a boiler and economizer unit. This is due to the fact that the rise in feedwater temperature through an economizer is greater for a low initial water temperature than for a high initial water temperature. Therefore, for a change of 100 deg. fahr. in feedwater temperature to the economizer, the change in feedwater temperature to the boiler will be somewhat less. As the change in superheat for a given rating depends on the change in feedwater temperature to the boiler, a given change in feedwater temperature to the unit will therefore cause a greater change in superheat for a boiler alone than for a boiler and economizer unit.

In the operation of the water-leg seal on the bleeder heater, it was found that there was water in the seal at all times, but that when this water flashed into steam further along in the piping the volume was increased to such an extent that a large pressure drop resulted in the pipe leading from the heater to the condenser. This pressure drop, which is the opposite action to that of an air lift, limited the amount of steam which could be bled, but as the amount of bled steam obtained was sufficient, no change was made. If more bled steam was desired, the bled-steam condensate would have to be cooled to more nearly the temperature of the condenser, or a small pump used to raise the pressure of this condensed bled steam to a point sufficient to put it in with the water leaving the hotwell of the main condenser (or still further along in the condensate system).

No. 1988

A GRAPHICAL STUDY OF JOURNAL LUBRICATION (PART III)[1]

By H. A. S. Howarth,[2] Philadelphia, Pa.
Member of the Society

This paper continues the investigation of journal lubrication reported to the Society under the same title in 1923 and 1924. Friction curves are here presented for central and offset partial bearings whose curvature radius exceeds that of the journal. The characteristics of fitted partial bearings are then studied, including their carrying capacities and friction.

THE second section of this paper dealt with the loading and displacement characteristics of offset and central partial bearings for which the journal radius of curvature was less than that of the bearing by an amount η. To complete their study there remained the determination of the friction coefficients, which will form the first part of the present paper.

2 In W. J. Harrison's paper[3] the fundamental equation for the friction at the surface of the journal is given as

$$f = \frac{\mu U}{h} + \frac{h}{2}\frac{\partial p}{\partial x}$$

which upon substitution of $dx = ad\theta$ becomes

$$f = \frac{\mu U}{h} + \frac{h}{2a}\frac{\partial p}{\partial \theta} \quad \ldots\ldots\ldots [15]$$

In this equation f is the frictional tangential force per unit area at the surface of the journal, opposing its rotation. Substituting

[1] Part I was presented in December 1923, and published in vol. 45 of the Transactions, p. 421. Part I contains Figs. 1 to 19, inclusive, and Equations [1] to [12]. Part II was presented in December 1924, and appears in vol. 46 of the Transactions, p. 809. Part II contains Figs. 20 to 34, inclusive, and Equations [13] and [14].

[2] General Manager, Chief Engineer, Kingsbury Machine Works.

[3] The Hydrodynamical Theory of Lubrication, Trans. Philosophical Society, Cambridge, England, vol. xxii, No. iii, pp. 39-54.

Contributed by the Special Research Committee on Lubrication and presented at the Annual Meeting, New York, November 30 to December 4, 1925, of The American Society of Mechanical Engineers.

in [15] the values of h from [1] [1] and $\frac{dp}{d\theta}$ from [14] [2] a useful formula for f is obtained.

$$f = \frac{\mu U}{2\eta}\left[\frac{1+4c\cos\theta-3c\cos\theta_1}{(1+c\cos\theta)^2}\right] \quad . \quad . \quad . \quad . \quad . \quad [16]$$

The total friction F at the journal surface for a given partial bearing is found by integrating $fad\theta$ between the limits of θ_4 and θ_3. See Fig. 20 (reproduced from Part II).

$$F = \int_{\theta_3}^{\theta_4} afd\theta \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [17]$$

3 This integration is readily performed mechanically for assumed values of c and θ_1, by plotting f against an angle base as

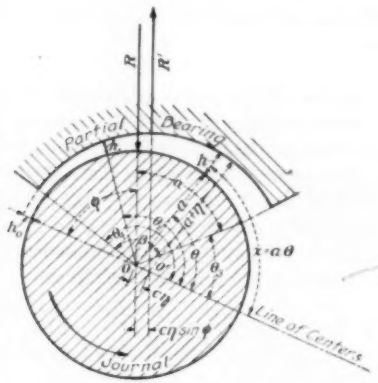


FIG. 20 SECTION OF JOURNAL AND PARTIAL BEARING ILLUSTRATING NOTATION USED IN FORMULAS

in Fig. 35. The area between vertical lines through θ_3 and θ_4 will be proportional to the friction force F exerted upon the journal surface by a bearing whose arc β extends from θ_3 to θ_4. Fig. 35 was drawn for $\theta_1 = 130$ deg. and $c = 0.4$ so as to correspond with Fig. 25 (reproduced from Part II) which shows the pressure curves for the same bearing. Thus for Figs. 25 and 35 $\beta = 190$ deg., $\alpha = 98.2$ deg., $\theta_3 = 17.2$ deg., $\theta_4 = 190$ deg., and $\phi = 74.6$ deg.

4 The journal friction coefficient λ for a given bearing is the ratio of the friction force F to the resultant pressure R, each of

[1] [1] $h = \eta(1+c\cos\theta)$

[2] [14] $\frac{dp}{d\theta} = \frac{6\mu U a}{\eta^2}\left[\frac{c(\cos\theta-\cos\theta_1)}{(1+c\cos\theta)^2}\right]$

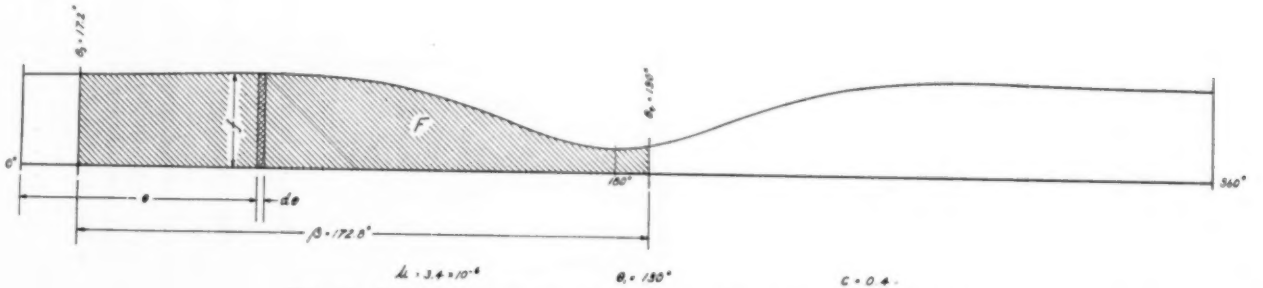


FIG. 35 FRICTION CURVE FOR THE PARTIAL BEARING SHOWN IN FIG. 25

which has been determined graphically for the same bearing in Figs. 25 and 35. λ is found to be directly proportional to η/a. The method above outlined was used to determine the friction coefficients for partial bearings of both the offset and the central classes.

5 Fig. 36 is the same as Fig. 30 of Part II except that the journal friction coefficient curves have been added. Fig. 37 is the same as Fig. 34 of Part II but with the journal friction coefficient curves added. In Fig. 37 there are two friction curves for $\beta = 180$ deg. One gives the coefficients for the 180-deg. *bearing*, while the other gives the coefficients for the *journal* running in a 180-deg. bearing. For all other values of β the journal friction alone is given.

6 In Fig. 38, the characteristics of the *full bearing* have been redrawn as in Fig. 10, except that the friction coefficient curve for the *bearing* has been added.

7 Before solving any examples from the above figures it is desirable to investigate another series of bearings herein termed *fitted partial bearings*.

Fitted Partial Bearings

8 These are bearings in which the journal and the bearing have the same radius of curvature. Using the same notations as heretofore, $\eta = 0$ and a is the radius of both the journal and the bearing. The journal will not run in a fitted bearing unless eccentric a

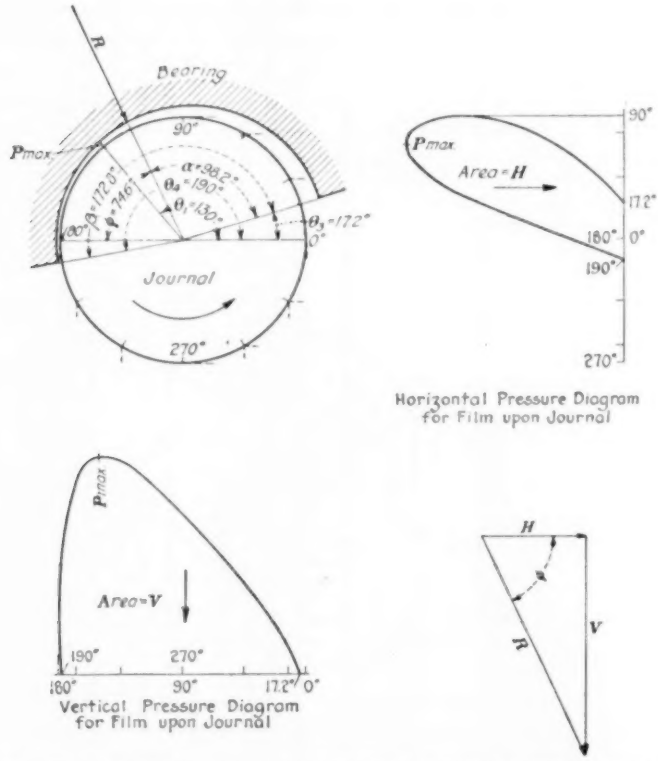


Fig. 25 Capacity Diagrams for 172.8-Deg. Partial Bearing ($c = 0.4$)

distance ε which depends upon the bearing arc β, the load, speed, and other factors that will appear below.

9 A typical relative running position of a fitted partial bearing and its journal is shown in Fig. 39. This is best studied if angular measurements are taken from OA, a line perpendicular to the line of centers OO'. The film thickness h is then expressed

$$h = \varepsilon \sin \theta \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [18]$$

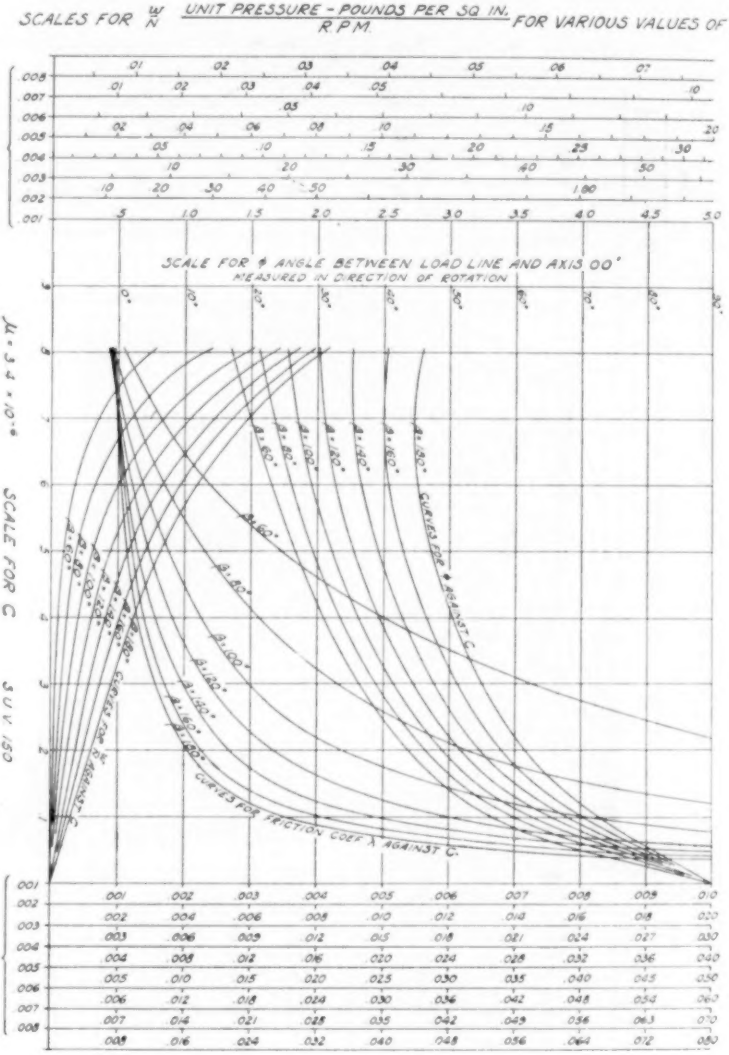

SCALES FOR W/N UNIT PRESSURE - POUNDS PER SQ. IN. / R.P.M. FOR VARIOUS VALUES OF l/d
SCALE FOR φ ANGLE BETWEEN LOAD LINE AND AXIS OO'
MEASURED IN DIRECTION OF ROTATION
SCALE FOR C
SCALE FOR λ, COEFFICIENT OF JOURNAL FRICTION.

FIG. 36 CHARACTERISTICS OF CENTRAL PARTIAL BEARINGS, WITH RADIAL RUNNING CLEARANCE ψ

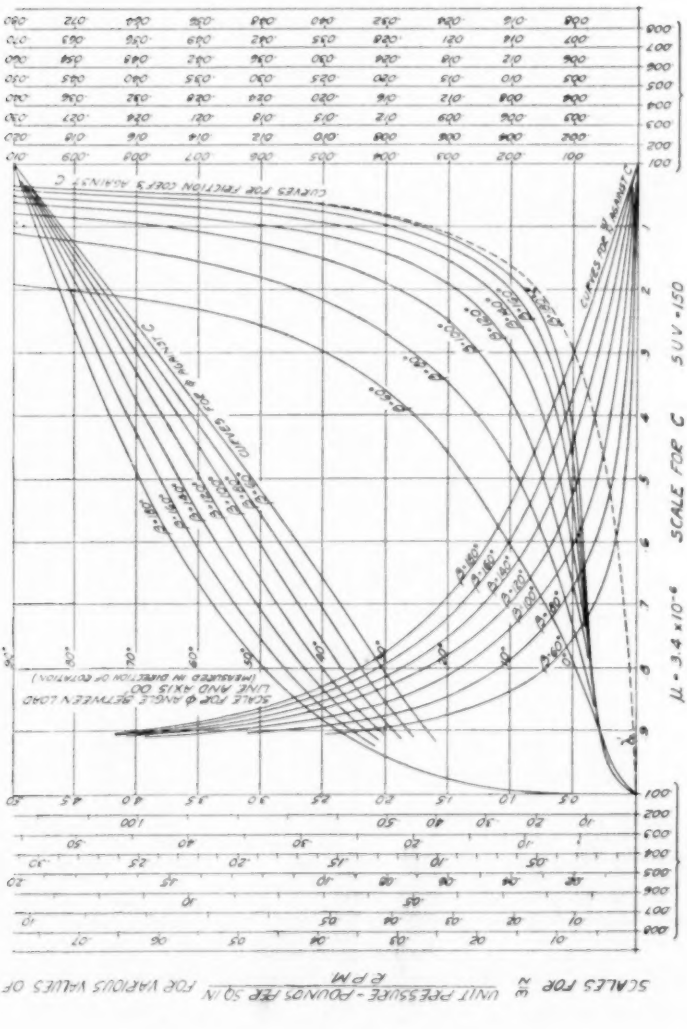

SCALE FOR λ, COEFFICIENT OF JOURNAL FRICTION
CURVES FOR FRICTION CODES AGAINST C
CURVES FOR ε AGAINST C
CURVES FOR ε AGAINST C
SCALE FOR ϕ ANGLE BETWEEN LOAD LINE AND AXIS OF ECCENTRICITY (MEASURED IN DIRECTION OF ROTATION)
μ - 3.4 x 10⁻⁶ SCALE FOR C SUV - 150
SCALES FOR w/N UNIT PRESSURE - POUNDS PER SQ IN / R P M FOR VARIOUS VALUES OF r/a

FIG. 37 CHARACTERISTICS OF OFFSET PARTIAL BEARINGS, WITH RADIAL RUNNING CLEARANCE 7

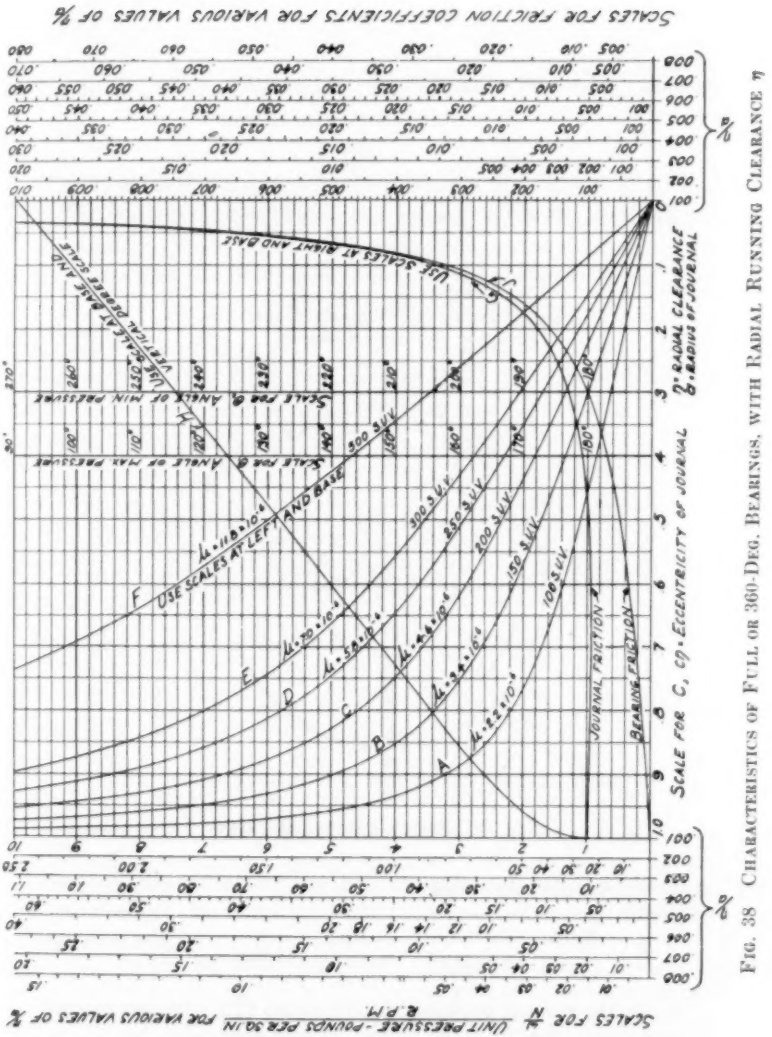

SCALES FOR FRICTION COEFFICIENTS FOR VARIOUS VALUES OF n/d
USE SCALES AT RIGHT AND BASE
USE SCALES AT LEFT AND VERTICAL PERIPHERAL SCALE
SCALE FOR θ, ANGLE OF MIN PRESSURE
ANGLE OF MAX PRESSURE
USE SCALES AT LEFT AND BASE
SCALE FOR θ, ANGLE OF MAX PRESSURE
300 S.U.V.
350 S.U.V.
200 S.U.V.
150 S.U.V.
100 S.U.V.
JOURNAL FRICTION
BEARING FRICTION
SCALE FOR C, eq. ECCENTRICITY OF JOURNAL
C, RADIAL CLEARANCE
r, RADIUS OF JOURNAL
SCALES FOR μN/P UNIT PRESSURE - POUNDS PER SQ IN FOR VARIOUS VALUES OF r/c
R.P.M.

FIG. 38 CHARACTERISTICS OF FULL OR 360-DEG. BEARINGS, WITH RADIAL RUNNING CLEARANCE π

The formula for pressure variation is readily obtained by taking again from Harrison's paper the fundamental relation

$$h^3 \frac{\partial p}{\partial x} = 6\mu U(h - h_1) \quad \ldots\ldots\ldots [13]$$

Assuming $\partial x = a\partial\theta$ we have the desired relation

$$\frac{\partial p}{\partial \theta} = \frac{6\mu U a(\sin\theta - \sin\theta_1)}{c^2 \sin^3\theta} \quad \ldots\ldots\ldots [19]$$

In Part II of this graphical study a similar equation was integrated mechanically, requiring much labor. Fortunately Equation [19]

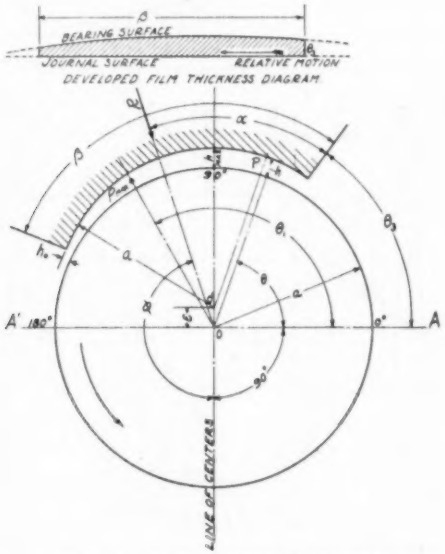


FIG. 39 FITTED PARTIAL BEARING

is readily integrated mathematically, and yields the unit pressure relation

$$P = \frac{6\mu U a}{c^2}\left[\frac{\sin\theta_1 \cos\theta}{2\sin^2\theta} - \frac{\sin\theta_1}{2}\log\tan\frac{\theta}{2} - \cot\theta\right] . \quad [20]$$

10 In the manner explained for Equation [16] the unit journal friction force f for the fitted partial bearing is found to be

$$f = \frac{\mu U}{c}\left(\frac{4\sin\theta - 3\sin\theta_1}{\sin^2\theta}\right) \quad \ldots\ldots [21]$$

11 From [20], by assuming θ_1, a pressure curve can be obtained. A series of such curves for several values of θ_1 is shown in

Fig. 40, plotted vertically from a diameter upon which the corresponding angular positions have been projected, from 0 deg. to 180 deg.

12 One of these curves, that for $\theta_1 = 120$ deg., is shown again in Fig. 41, to an enlarged vertical scale, by line AA'. The circle represents the journal which is assumed to rotate counter-clockwise. In this figure the eccentricity of the journal to the bearing is assumed to take place along the radius from the center C to 90 deg., as indicated in Fig. 39. The sectioned portion in Fig. 41 represents the part of the journal surface upon which the partial fitted bearing may lie. Obviously the fit must be *less* than 180 deg. if the journal is to have any freedom to assume a running position that will permit the formation of a wedge-shaped oil film.

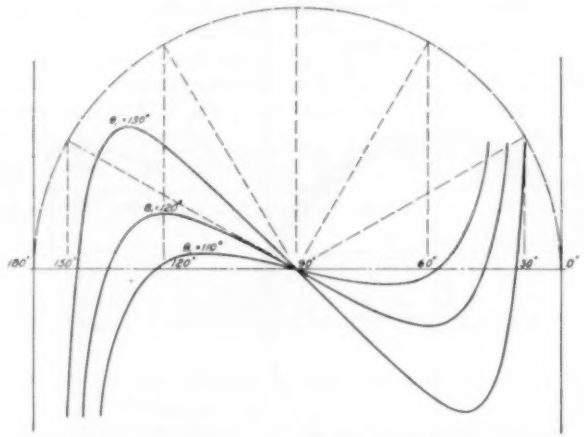


Fig. 40 Pressure Curves for Fitted Partial Bearings

The line BB' at the right of the journal shows the pressure curve laid off from angles projected upon the 0 deg. to 90 deg. radius. This figure may now be used in the same manner as previously described, to determine resultant pressures for bearings of various angular lengths.

13 In Fig. 41 draw lines FGH and $F'G'H'$ at any distance x from and parallel to the horizontal and vertical pressure base lines 0 deg.-180 deg., and 0 deg.-90 deg., respectively. The letters indicate the intersections with the pressure curves. Project these intersections upon the journal surface to the points $F''G''H''$. The interpretation of the result obtained is that two partial bearings may be had, both terminating at F'' and having the point of maximum pressure at $D''(\theta_1 = 120$ deg.). One of these bearings would start at G'' and have all positive pressures in its film, whereas the other would start at H'' and have negative pressures as

far as G'' and positive pressures the rest of the way to F''. The areas FDG and GEH represent positive and negative vertical carrying capacities, respectively. The area FDG is the capacity of partial bearing $F''G''$ in the vertical direction. Similarly area FDG minus area GEH is the vertical capacity of partial bearing $F''G''H''$. The horizontal capacities are likewise found from curve BB' to be $F'D'G'$, and $F'D'G'$ minus $G'E'H'$. Evidently the magnitude and direction of the resultant pressure upon each of these

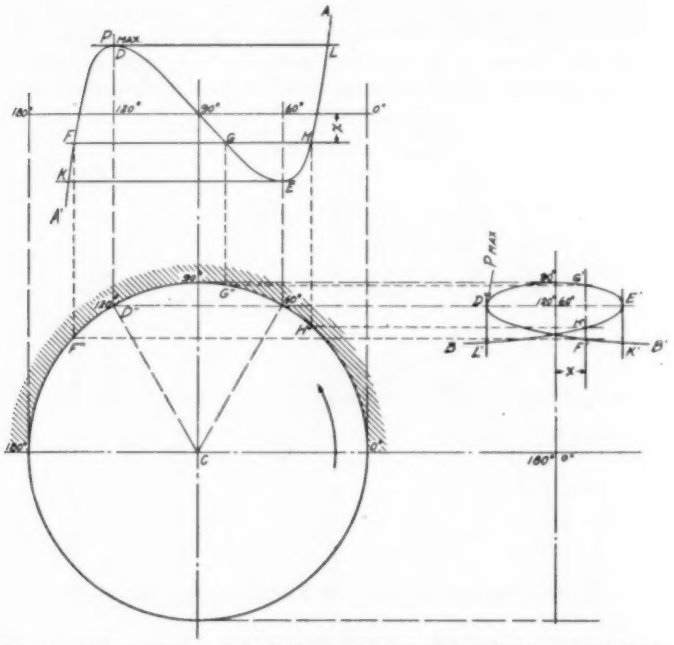


FIG. 41 HORIZONTAL AND VERTICAL PRESSURE COMPONENTS FOR FITTED PARTIAL BEARINGS

two partial bearings is obtainable by mechanical integration of the areas above mentioned. Each resultant R will make an angle ϕ with the line of centers OO' as indicated in Fig. 39, measuring in direction of rotation from the resultant.

14 From Equation [**20**] for the pressure P it is evident that the resultant pressure R will have as a factor in it $(\mu U a^2/\varepsilon^2)$, the extra a factor coming in through integration. R is therefore directly proportional to the oil viscosity μ, and the surface speed U. It is inversely proportional to the ratio $(\varepsilon/a)^2$. Hence the closer the relative approach of the bearing surfaces the greater will be the carrying capacity of the film.

15 The tangential friction force on the journal running in a partial bearing is as above explained $F = a\int_{\theta_3}^{\theta_4} f d\theta$. The unit friction force f is found from [**21**]. If it be laid off at right angles to a base line representing θ the areas under the curve will be proportional to the friction F between whatever limits are chosen. Using the same data for both figures, the angle θ_3 for the bearing $H''F''$ of Fig. 41 will lie at H in the friction diagram of Fig. 42, while for bearing $G''F''$ θ_3 will lie at G. Angle θ_4 will lie at F for both bearings. Area $HFCA$ in Fig. 42 is proportional to the total tangential journal friction for bearing $H''F''$, whereas area $GFCB$ similarly represents it for bearing $G''F''$.

16 In the friction force F there appears the factor $\mu Ua/c$, the a entering by integration. Hence when the coefficient of friction λ is determined from $\lambda = F/R$ it will have the factor c/a in it, the factors μ and U cancelling out.

17 What was found from Figs. 41 and 42 to be true of the pressure and friction relations for bearings having their maximum pressure at $\theta_1 = 120$ deg., can be found similarly for any other value of θ_1. The resultant R will vary not only for every value of x, but also with the assumed value of θ_1. For every assumption made for θ_1 there may be obtained a series of partial bearings for which all the following factors will be known:

(*a*) Where the partial bearing starts with respect to the line of eccentricity
(*b*) Where the partial bearing ends
(*c*) Where the resultant pressure lies with respect to the bearing
(*d*) Where the resultant pressure lies with respect to the line of eccentricity
(*e*) The magnitude of the resultant in terms of viscosity, speed and relative eccentricity
(*f*) The friction coefficient for the journal, with respect to the eccentricity of the partial bearing.

18 How the above factors vary with relation to each other will be shown from the following figures. Instead of using the resultant R and the surface speed U it is more convenient to use the pressure w per unit of nominal projected area, and the revolutions per minute N.

19 The resultant $R = 2wa$ and $U = (2\pi aN/60)$. Hence the expression for $R = \dfrac{\mu Ua^2}{c^2}$ [function of θ] can be changed to $w = \dfrac{\pi N\mu}{60(c^2/a^2)}$ [function of θ] from which is obtained the relation

$$\frac{w}{N} = \frac{\pi\mu}{60(c/a)^2} \text{ [function of } \theta\text{]} \ldots\ldots \text{ [22]}$$

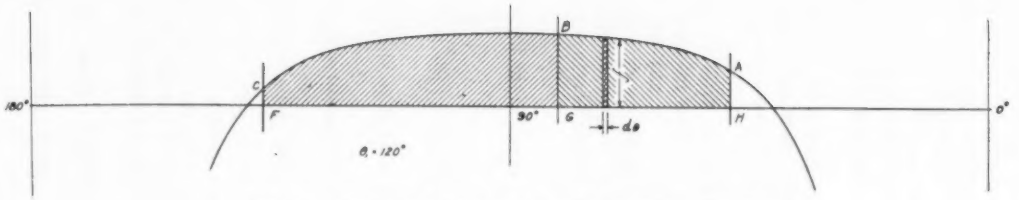


FIG. 42 FRICTION CURVE FOR FITTED PARTIAL BEARING

w/N is therefore *directly* proportional to the viscosity μ and *inversely* proportional to the square of the ratio of the eccentricity to the radius of the journal $(\varepsilon/a)^2$.

20 In order to obtain data for useful charts, figures similar to Figs. 41 and 42 were drawn for numerous angles of θ_1 and for $\varepsilon/a = (0.001)$ and $\mu = (3.4 \times 10^{-6})$. Upon each of these a series of lines similar to FGH was laid off for definite values of β. For example the line FG or line FH in Fig. 41 was chosen as some definite angular length β, and its exact location x determined. The angles θ_3 and θ_4 were then measured. The corresponding $F'G'$ or $F'H'$ line would then be drawn and the resultant pressure determined as above explained. Having predetermined θ_1 and β throughout a chosen range, the corresponding values of R, α, ϕ and λ were then found. For purpose of plotting w/N is used in lieu of R.

21 The characteristics of these fitted partial bearings can best be understood by referring to Fig. 43 in which α is plotted against w/N, assuming $\varepsilon/a = 0.001$ and $\mu = 3.4 \times 10^{-6}$. α is the angle the resultant pressure makes with the leading edge of a bearing that is β degrees long. Hence the ratio α/β shows relatively where the resultant pressure intersects the bearing surface.

22 Entering the base of Fig. 43 at A where $\alpha = 56$ deg., a vertical line will cut the curve for $\beta = 90$ deg. at two points, B and C. There are therefore two conditions under which a 90-deg. bearing can have a leading angle of 56 deg. Reading on the scale at the left we find intercepts OD and OE representing their carrying capacities. Remembering that w/N is proportional to $(\varepsilon/a)^2$, we realize that if the eccentricity is suitably decreased the capacity of the bearing at B will be the same as that at C for which $\varepsilon/a =$

0.001. Similarly the capacity of bearing C, if the eccentricity be sufficiently increased, will be the same as that of the bearing B for which $\varepsilon/a = 0.001$.

$$\left(\frac{w}{N}\right)_B = K\,\frac{AB}{\left(\frac{\varepsilon}{a}\right)^2_b} \qquad \left(\frac{w}{N}\right)_C = K\,\frac{AC}{\left(\frac{\varepsilon}{a}\right)^2_c}$$

Hence if $(w/N)_B = (w/N)_C$ we have

$$\left(\frac{\varepsilon}{a}\right)_B = \left(\frac{\varepsilon}{a}\right)_C \sqrt{\frac{AB}{AC}} \quad \ldots\ldots\ldots \quad [23]$$

23 If, therefore, the bearing can run in either of these two positions, ε_B or ε_C, which will it choose? Evidently the one for which the *friction* is the least. This can be shown to be the upper

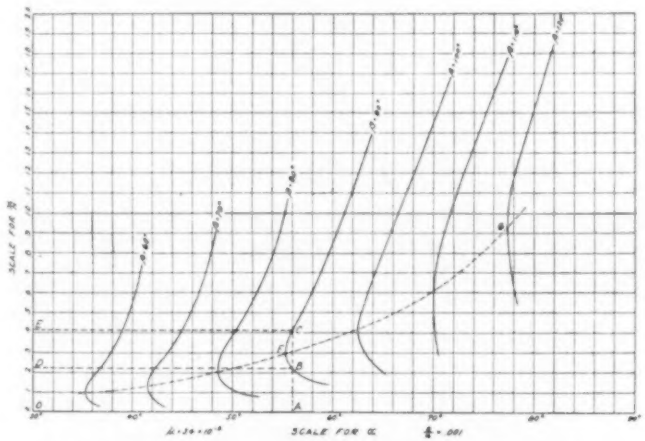


FIG. 43 CAPACITY CHART FOR FITTED PARTIAL BEARINGS

position C. But if the bearing is running in position B how can it rise to position C? Evidently α or β must change. It is found that bearing B has a vacuum at its leading edge. The place where the bearing starts can therefore change so that with continually decreasing friction and increasing eccentricity the angles α and β will at first decrease and then increase until the position C is reached.

24 There is only one angle α for each bearing β where two such points as B and C would coincide. That is the tangent one F for which $\alpha = 55.2$ deg. for a bearing 90 deg. long. There is a series of points similar to F, covering the whole range of values of β. The ratio α/β is not the same for the series, although it varies

only a little for a wide range of β. This minimum α/β is considerably above 0.5 for the usual bearing proportions. Further comments will be made on this fact later.

25 The question now arises as to what series of partial fitted bearings shall be chosen for further study. Referring again to Fig. 41, the lowest possible position of line *FGH* would be *KE* and the highest *DL*. The intercept *FE* would, as the line is lowered from *D*, increase from zero at *D* to a maximum at *KE*. The area *KDE*

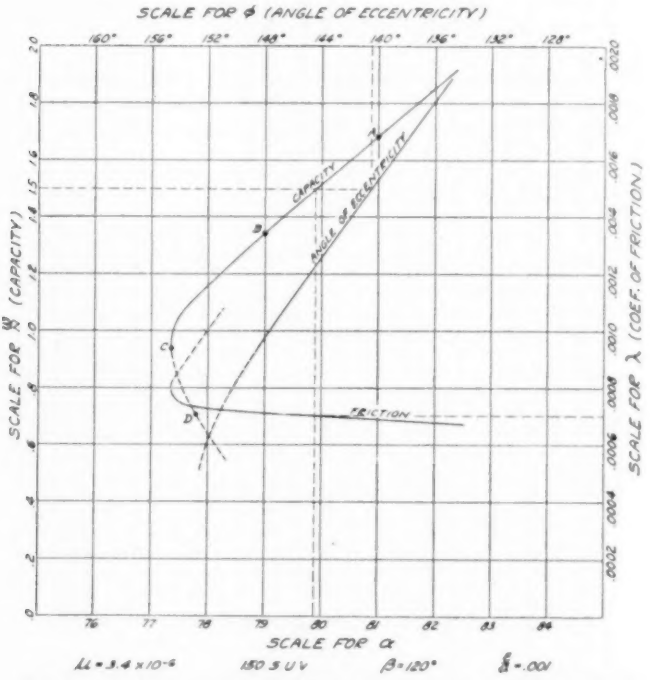

SCALE FOR ϕ (ANGLE OF ECCENTRICITY)
160° 156° 152° 148° 144° 140° 136° 132° 128°
SCALE FOR W (CAPACITY)
SCALE FOR λ (COEF. OF FRICTION)
CAPACITY
ANGLE OF ECCENTRICITY
FRICTION
SCALE FOR α
76 77 78 79 80 81 82 83 84
μ = 3.4 x 10⁻⁶
150 S U V
β = 120°
δ = .001

FIG. 44 CHARACTERISTICS OF 120-DEG. FITTED PARTIAL BEARING

gives the bearing of maximum carrying capacity, because the area is wholly positive. To increase the intercept β it is necessary to raise the line *KE* and take an intercept like *FGH*. This at once introduces a negative pressure in the portion of the bearing β between *G* and *H*. The bearing of maximum capacity, *KDE* combined with *K'D'E'*, in Fig. 41 would produce a point similar to *G* on curve β = 120 deg. on Fig. 43.

26 In Fig. 44, the characteristics of the series of 120-deg. bearings are shown, including the leading angles α, the directions of

eccentricity ϕ, and the friction coefficients λ. The w/N scale is drawn for $\mu = 3.4\times10^{-6}$ and $\epsilon/a = 0.001$.

EXAMPLE 7. Investigate fitted 120-deg. partial bearings carrying a unit load of $w = 75$ lb. per sq. in. at a speed of $N = 50$ r.p.m. on oil having a viscosity of 150 S.U.V. A series of points A, B, and C, are chosen at random in Fig. 44 on the w/N curve, each representing a different bearing. Their leading angles are found to be $\alpha = 81$ deg., 79 deg., and 77.35 deg., respectively. The corresponding values of ϕ are next determined, and are found to be

$$\phi_A = 137.8 \qquad \phi_B = 142.8 \qquad \phi_C = 148.5$$

These give the directions of eccentricity. To determine the eccentricity and the friction coefficients and the minimum film thickness for each proceed as follows. $w/N = 1.5$. Hence the corresponding values of ϵ/a are found as above explained to be

$$\left(\frac{\epsilon}{a}\right)_A = 0.001\sqrt{\frac{1.68}{1.5}} = 0.00106$$

$$\left(\frac{\epsilon}{a}\right)_B = 0.001\sqrt{\frac{1.34}{1.5}} = 0.000945$$

$$\left(\frac{\epsilon}{a}\right)_C = 0.001\sqrt{\frac{0.94}{1.5}} = 0.000792$$

The corresponding friction coefficients are found to be

$$\lambda_A = 0.00069\times0.00106 = 0.000732$$
$$\lambda_B = 0.00071\times0.000945 = 0.000670$$
$$\lambda_C = 0.0008 \times0.000792 = 0.000633 \text{ min.}$$

27 Applying the above data as in Fig. 45 it is seen how these three bearings differ from each other, assuming the resultant pressure vertical in each case. The minimum film thickness h_0 would from Equation [**18**] be $\epsilon \sin(\theta_3+\beta)$, which can be changed to the following more convenient form

$$\left(\frac{h_0}{a}\right) = \left(\frac{\epsilon}{a}\right)\cos\psi \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [\mathbf{24}]$$

in which

$$\psi = 180-(\alpha+\phi-\beta)$$

For the three bearings A, B and C of Fig. 45 we find

$$(h_0/a)_A = 0.00106 \cos 81.2 \text{ deg.} = 0.000162$$
$$(h_0/a)_B = 0.000945 \cos 78.2 \text{ deg.} = 0.000193$$
$$(h_0/a)_C = 0.000792 \cos 74.15 \text{ deg.} = 0.000216$$

It may be thought that a point D below C on the w/N curve may carry our load with less friction than C. Taking such a point D on the w/N curve, in Fig. 44, for which $\alpha = 77.8$, we find $(w/N)_D = 0.70$ on the scale for $(\epsilon/a) = 0.001$. Our problem states that $(w/N) = 1.5$. Hence the actual eccentricity for D would be $\left(\frac{\epsilon}{a}\right)_D = 0.001\sqrt{\frac{0.70}{1.5}} = 0.000683$. The coefficient of friction would therefore be $\lambda_D = 0.00094\times0.683 = 0.000642$, which is *higher* than for point C.

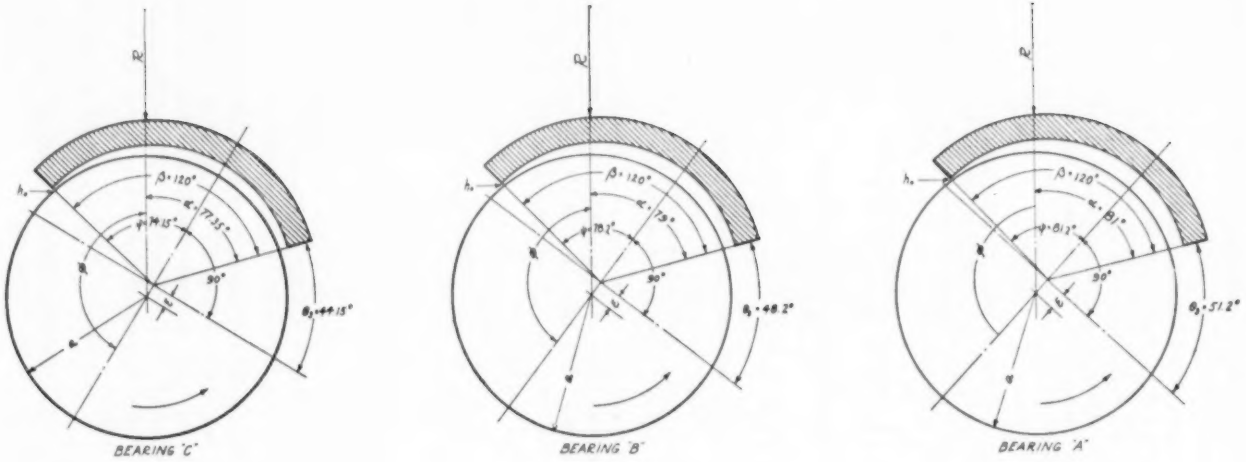


FIG. 45 THREE 120-DEG. FITTED PARTIAL BEARINGS

$\frac{\omega}{N} = 1.5 \quad \mu = 3.4 \times 10^{-6}$

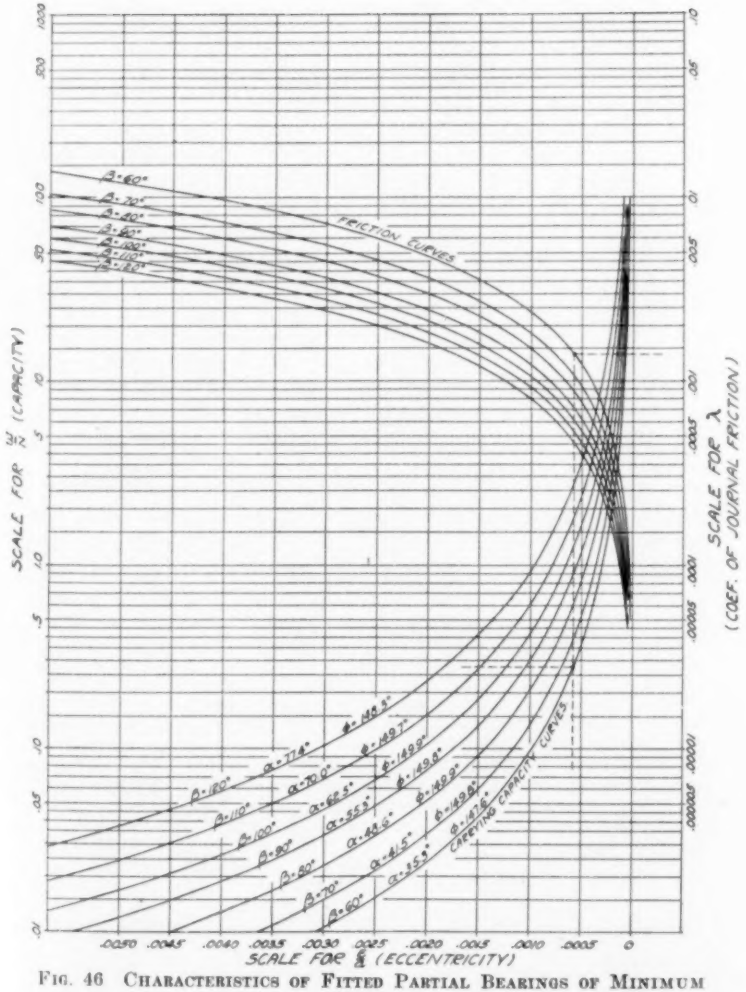

FRICTION CURVES
CARRYING CAPACITY CURVES
SCALE FOR $\frac{W}{N}$ (CAPACITY)
SCALE FOR λ (COEF. OF JOURNAL FRICTION)
SCALE FOR $\frac{e}{c}$ (ECCENTRICITY)

FIG. 46 CHARACTERISTICS OF FITTED PARTIAL BEARINGS OF MINIMUM FRICTION

$\mu = 3.4 \times 10^{-6}$

28 It therefore appears that when the 120-deg. bearing is carrying a load $w/N = 1.5$, the value of α for the greatest h_0 (minimum film thickness) and for the minimum friction, lies at the point of inflection C on the w/N curve in Fig. 44, for which $\alpha = 77.35$ deg. for this 120-deg. bearing. The corresponding minimum film thickness is $h_0 = 0.000216\ a$ for an oil with a viscosity of 3.4×10^{-6} (S.U.V. = 150).

29 The above example has shown that fitted partial bearings have minimum friction when α/β has but one value for each angle β. The dotted curve through F and G in Fig. 43 covers this series of bearings. The β scale could be laid off on it, 90 deg. falling at F, 120 deg. at G, and so on.

30 The properties of this point-of-inflection series of bearings have therefore been charted in Fig. 46 for $\beta = 60$ deg., 70 deg., 80 deg., 90 deg., 100 deg., 110 deg., and 120 deg.; ϵ/a has been chosen as the base to make the solving of bearing problems easier. Correction need only be made for viscosity, and this is done as explained in the following examples. For each bearing angle β the corresponding angles α and ϕ are marked with it along the w/N curves.

31 The charts for solving bearing problems are now in convenient enough form for ready use. See Figs. 36, 37, 38, and 46.

32 By solving a problem for all four types of bearings their comparative characteristics will be evident. Examples 3, 4, and 5, in Part II, may be supplemented and summarized by Example 8 herewith.

EXAMPLE 8. A 2-in. journal runs at 300 r.p.m. carrying a unit load of 60 lb. per sq. in. of projected area, i.e., 120 lb. per inch of axial length. Find characteristics for this journal when running with radial clearance of 0.002 in. in a full bearing, in a central, in an offset, and in a fitted 120-deg. partial bearing. Assume viscosity is 3.4×10^{-6} (150 S.U.V.). For some of the characteristics see Examples 3, 4, and 5.

$w/N = 60/300 = 0.2$. Hence from Fig. 36 $\lambda = 0.0022$ while from Fig. 37 $\lambda = 0.0019$. This completes the data for all cases except the fitted partial bearing which will be studied below.

The fitted 120-deg. bearing can be studied from Fig. 46. Entering the w/N scale with 0.2 we find from the 120-deg. capacity curve that $\alpha = 77.4$ deg., $\phi = 148.3$ deg. and that $\epsilon/a = 0.00215$, $\lambda = 0.0017$, and $(h_0/a) = (\epsilon/a)\cos\psi$.

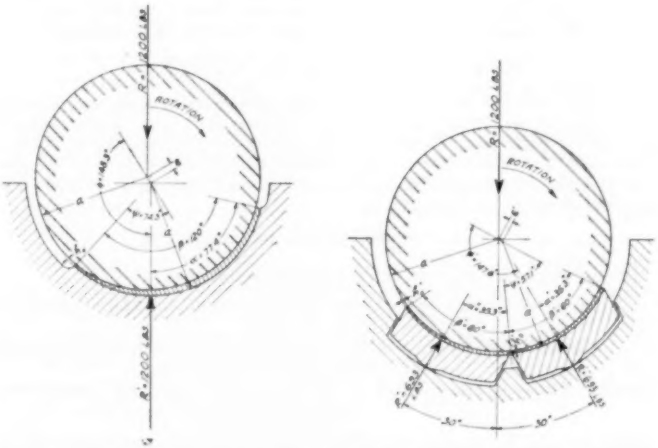
$$\psi = 180-(77.4+148.3-120) = 74.3 \text{ deg.}$$
$$(h_0/a) = 0.00215\times0.2706 = 0.000582$$

33 The data on these four bearings can now be tabulated for comparison:

	β deg.	α deg.	ϕ deg.	c, c_η, ϵ	h_0, h_n	λ	Data from example No.
I Full bearing	360	....	90	$c = 0.235$ / $c\eta = 0.00047$	0.00153	0.00325	3
II Offset partial ...	120	75.6	51.9	$c = 0.638$ / $c\eta = 0.001276$	0.000735	0.0019	4, 8
III Central partial ..	120	60	31.8	$c = 0.642$ / $c\eta = 0.001284$	0.000716	0.0022	3, 8
IV Fitted partial ...	120	77.4	148.3	$\epsilon = 0.00215$	0.000582	0.0017	8

34 Thus by comparing these bearings it is found that the *journal friction* coefficient is lowest for the fitted, next for the offset, higher for the central, but highest of all for the full bearing. The distance of closest approach does not quite correspond, being least for the fitted, next for the central, greater for the offset, and greatest of all for the full bearing.

35 It must be remembered, however, that the clearance ratio was assumed to be $\eta/a = 0.002$. This ratio is not the one for the minimum friction of bearings I, II, and III. In bearing I, as was learned from Example 6, the best clearance ratio would be somewhere between 0.002 and 0.003 and the minimum friction coefficient would be about 0.0031.



FIGS. 47 AND 48 COMPARISON OF 120-DEG. FITTED PARTIAL BEARING AND THE EQUIVALENT DOUBLE PIVOTED-SEGMENT BEARING

(See Example 9.)

36 The study of partial bearings has now proceeded far enough to answer intelligently an important question that occasionally arises in bearing designing, namely, "Is there a saving in friction if two or more pivoted segments are used instead of a single partial bearing to support a journal?" This question can most definitely be answered by assuming both types of bearings to cover the same arc of the journal's surface. In Fig. 47 a fitted 120-deg. partial bearing is shown, while in Fig. 48 two pivoted fitted 60-deg. segments are shown adjacent, their most favorable position.

37 In the single 120-deg. bearing the journal must lift and shift to the left so as to form the wedge-shaped film. The eccentricity, the distance of closest approach, and the frictional coefficient can be determined by referring to Fig. 46.

38 In the double segmental bearing the journal must lift and the segments must tip to form the wedge-shaped films. Similar data can be obtained for this bearing also from Fig. 46. Example 9 shows the single bearing to be much superior to the double pivoted segmental bearing. More than two segments would be inferior to two.

EXAMPLE 9. Compare the friction, eccentricity and nearest approach of surfaces for a journal 6 in. diameter, 6 in. long, running at 400 r.p.m., carrying a vertical load of 7200 lb., (*a*) when supported in a 120-deg. fixed fitted bearing, and (*b*) when supported in two adjacent 60-deg. fitted pivoted segments. Assume the oil viscosity $\mu = 3.4 \times 10^{-6}$ (inches, pounds, seconds). The vertical load will be 7200/6 = 1200 lb. per inch of axial length of the bearing.

(*a*) The nominal unit pressure $W_a = 1200/6 = 200$. Hence $W_a/N = 0.5$. Entering Fig. 46 with this value we find $(\epsilon/a)_a = 0.0014$ and $\lambda_a = 0.0011$. The nearest approach found from Formula [**24**] is found to be

$$(h_o/a)_a = 0.0014 \cos 74.3 \text{ deg.} = 0.00038 \text{ in.}$$

(*b*) The vertical load per linear inch (axial) is in this case divided, each segment carrying (1200/2) sec 30 deg. = 693 lb. Each segment must be studied separately to avoid confusion. Hence $W_b = 693/6 = 115.5$. $W_a/N = 115.5/400 = 0.29$. From Fig. 46 we find $(\epsilon/a)_a = 0.0006$ and that the friction coefficient $\lambda_b = 0.00135$ sec 30 deg. = 0.00156. The nearest approach is found to be

$$(h_o/a)_b = 0.0006 \cos 57.1 \text{ deg.} = 0.000326$$

(*ab*) Comparing the two bearings it is found that

Friction coefficient $\lambda_b = 1.42\lambda_a$ markedly favoring the single bearing

Eccentricity $(\epsilon/a)_b = 0.43(\epsilon/a)_a$ markedly favoring the single bearing

Nearest approach $(h_o/a)_b = 0.86(h_o/a)_a$ also favoring the single bearing.

39 It was explained in Parts I and II of this graphical study that it was limited to the two-dimensional theory, and that the end-leakage effect was left out of account. Another important assumption was that the oil viscosity μ within the film was constant from one end of the film to the other.

40 This constant-viscosity assumption, while reasonably true for a slow-speed bearing or a bearing with a water-cooled shell, would not be reasonable for a high-speed bearing to which fresh oil is steadily supplied at the entering end of the oil film and steadily discharged at a higher temperature at the other end. If the viscosity falls off as the oil passes through the film the pressure curve will change, thereby altering the location of the resultant pressure R, the probable tendency being to reduce the value of the ratio α/β in partial bearings. The exact law of viscosity variation would have to be known in order to obtain an accurate result, true for a given case. Two assumptions, each extreme, might be made. One is that the viscosity is constant. The other is that all of the heat of friction is absorbed by the oil in the film

and discharged with it. It is quite probable that the graphical method will offer a ready means for at least an approximate solution of bearing problems, assuming various rates of viscosity change through the lubricating film.

41 This graphical analysis of journal lubrication has been made possible by the generous encouragement of Mr. Albert Kingsbury, without whose kind coöperation the necessary time would not have been available. For the accuracy of the computations, curves, and charts the author is largely indebted to the able assistance of Mr. Nelson Ogden and Mr. S. J. Needs, engineers on Mr. Kingsbury's staff.

DISCUSSION

GEORGE B. KARELITZ.[1] This paper gives for the first time numerical values of the friction in bearings. The diagrams presented bring the hydrodynamical formulas of friction to a shape from which it is easy to draw conclusions.

In applying the given data for practical purposes, however, certain points ought to be taken into consideration. The present tendency is to make the bearings as short as possible, and the end leakage of the oil causes considerable departure from the conditions of a long bearing dealt with in the paper. The minimum friction occurs, for instance in Example 6, at a clearance higher than used in present practice. But the flow of oil along the bearing grows rapidly with the increase of the clearance. The large oil leakage from the ends will impair the carrying capacity of the oil film, and the thickness of the oil film will be smaller than that anticipated by the diagrams for a long bearing. The gain in friction thus may become doubtful.

The thickness of the oil film might be improved by increasing the amount of oil supplied to the bearing. In the case of forced-feed bearings this would mean a larger oil-pump capacity and probably higher oil pressures or larger pipe sizes. Thus we come to a question of heat or work balance of the oiling system. In the case of oil-ring-lubricated bearings, the amount of oil delivered to the journal is limited by the carrying capacity of the oil rings, thus limiting the allowable clearance. In general, however, the question of oil flow in and through the bearing is of high importance and investigators should exert their efforts along this direction, which so far has hardly been explored.

The coefficients of friction for full bearings are derived with the supposition of an oil inlet being near the minimum-pressure point. This seldom occurs in practice. If the inlet is on the plane normal to that of the load, the pressures of oil in the no-load half of the bearing would have a high negative value, which is impossible to conceive.

[1] Research Dept., Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa.

In the case of oil-ring sleeve bearings, the upper half of the bearing is cut through and sometimes grooved, so that the oil film is broken, and pressures cannot be built up. This will tend to make the oil film in the pressure part considerably thinner, increasing the friction in the bearing.

A minor remark on Equation [**20**] is that the constant of integration is omitted. The pressure at the start of the bearing being, say, P_0, the equation should be written as follows:

$$P = P_0 + \frac{6\mu Ua}{\epsilon^2}\left[\frac{\sin\theta_1\cos\theta}{2} - \frac{\sin\theta_1}{2}\log\tan\frac{\theta}{2} - \operatorname{cotan}\theta\right] - \frac{6\mu Ua}{\epsilon^2}\left[\frac{\sin\theta_1\cos\theta_3}{2} - \frac{\sin\theta_1}{2}\log\tan\frac{\theta_3}{2} - \operatorname{cotan}\theta_3\right].$$

H. T. NEWBIGIN.[1] In dealing with radial difference between the journal and the brass, the assumption is made that a constant ratio of running clearance to the shaft radius will give an effective-pressure oil film over an equal angular surface in two bearings of widely different diameters. It appears to the writer that for a given clearance per inch of shaft radius this angle will decrease as the shaft diameter increases, because the actual thickness of oil in which effective pressure can be generated at a given speed with a given oil is constant.

The theory does not take end leakage into account. This is an important omission, because improvement in journal-bearing design as regards reduction of length is desirable, and with such reduction of length the effect of end leakage becomes increasingly important.

The modern practice of designing bearings according to the Reynolds theory of lubrication originated with Beauchamp Tower's experiments on journal bearings, and it was shown that the pressure-oil-film effect was produced by reason of the running clearance. This resulted in the development of thrust bearings where the oil wedge is produced by a pivoted-pad construction. A developed section of a journal shows that such a bearing is merely a single pad bearing having a permanently backed-off surface, and thrust bearings now are being made with permanently backed-off segments instead of pivoted pads. Although the possibility of saving in friction in ordinary journal bearings is usually less than was the case with the older form of thrust bearings, there are inherent advantages in the pivoted-pad construction even when applied to journal bearings.

According to the author's theory, a two-pad bearing embracing an angle of 120 deg. is inferior to a fixed, fitted bearing embracing the same arc of contact. This is not borne out by the writer's experience. For example, in the case of four 10-in.-diameter bearings fitted to the necks of some cold rolls running at 30 r.p.m., and loaded to about 1000 lb. per sq. in. in which two-pad pivoted

[1] Michell Bearings, Ltd., Newcastle-on-Tyne, England.

bearings replaced fitted bearings, both embracing an arc of 120 deg., the saving in friction is from 75 to 90 per cent. The saving in this case is that between greasy friction and film lubrication, but even in cases where film lubrication obtains in bearings of the ordinary type, the reduction in friction is at least equal to the saving in length. The saving in length is usually the deciding factor. In the case of large bearings this is an important item, and is brought about without increasing the maximum unit pressure.

In Fig. 45, the author suggests the use of non-pivoted, fitted, offset, journal bearings, but the writer doubts whether in practice these would be found satisfactory, because the surfaces at the leading edge do not converge in the direction of motion, and are therefore incapable of generating the necessary wedge of oil to give eccentricity of the shaft to the brass. Furthermore, if such bearings were short axially in relation to their circumferential length the effect of end leakage would be considerable.

Michell's mathematical study, The Lubrication of Plane Surfaces, has enabled the problem to be solved for three dimensions. It applies equally to journal and to thrust bearings, and several hundred of the former bearings fitted with pivoted pads are justifying this method of construction.

The question of the number of pads in a journal is chiefly the practical one of insuring equality of loading. Where the load line on the journal bisects the radial reactions from the pivoting lines of two pads, the load is divided equally between them as shown by the author's Fig. 48.

ROBERT C. H. HECK.[1] The writer's interest in the quantitative detail of this subject is rather recent, and has taken the form of an effort to put the beginning of the theory into shape for presentation to students.

The starting point of all deductions of pressure variation within the film is the Reynolds primary differential equation

$$\frac{dp}{dx} = -6\mu U \frac{h-h_1}{h^3},$$

quoted as Equation [13] in Part II of this series.[2] This is established by making definite the general equation

$$\frac{dp}{dx} = \mu \frac{d^2u}{dy^2}$$

in which μ is the variant velocity across the film.

[1] Professor of Mechanical Engineering, Rutgers College, New Brunswick, N. J. Mem. A.S.M.E.

[2] Trans. A.S.M.E., vol. 46 (1924), p. 809. The same equation appears as the first equation in the paper entitled Charts for Studying the Oil Film in Bearings, by George B. Karelitz. See page 1101, this volume.

By keeping close to physical considerations and working within the restrictions of two-dimensional action in an oil film, this equation can be established by a much shorter and simpler line of reasoning than is to be found in the classic presentation. The latter first takes into account every possible component action, then throws away all but the few elements that belong to the restricted problem; it can be found, in clear and convenient shape, in *Engineering*, vol. 100 (1915), page 101. For the short derivation see the writer's Mechanics of Machinery, Kinematics and Dynamics, page 455.

Thus to develop Equation [**13**] in closer contact with the physical action that it represents has one decided advantage, in that the limitations of its applicability become much more clearly apparent. Some of the mathematical investigators do not seem to have appreciated the fact that this equation expresses only a component action, which is predominant and controlling in a portion of the film; in other parts it is one among several, all of the same general order of magnitude, of little strength and scarcely capable of close formulation.

Certainly this equation does not apply all around the circle of a complete bearing, which is the reason why no experiment has come anywhere near to realizing Harrison's first curve of pressure variation as reproduced at the beginning of Part I of the present series.

Fortunately the limited range of the partial bearing is that within which Equation [**13**] actually or nearly prevails, so that the useful portion of these papers has a good foundation. But due to the uncertainty of this fundamental theory, plus the neglect of end leakage or escape, they can be no more than guiding approximations. At that they come much nearer to making film theory a basis of design than any preceding workout along similar lines.

THE AUTHOR. The discussion of this paper has touched upon several important aspects of the journal-bearing problem that deserve amplification. Before dealing with them individually it appears best to examine a general case that will cover several phases.

The charts of full- and partial-bearing characteristics, Figs. 36, 37, 38, and 46, are based on the assumption that the oil film is continuous throughout the bearing arc in each case. It is possible, however, by their means to study the lubrication of bearings whose deficient oil supply prevents the formation and maintenance of complete films.

The study of a full bearing supplied with oil at a gradually reduced rate will cover the field for bearings provided with running clearance. The maximum oil supply required should be enough to make up side leakage, so as to maintain a complete film, and then,

if the speed is high enough to require it, additional oil should be passed across the film to keep the bearing at whatever temperature is desired. Oil to such a bearing may be supplied through one longitudinal groove, located at the point of minimum pressure. With the exception of this groove the bearing surface should be unbroken. For convenience the bearing width, axially, is assumed to be twice the journal diameter.

A gradual reduction of the oil-supply rate in such a high-speed bearing would be accompanied by a gradual increase in the bearing temperature, but by a decrease in the frictional resistance to rotation of the journal. An oil-supply rate would be reached at which the film would no longer remain complete. From this point on it may be assumed that the film will be perfect throughout a shorter arc from 360 deg. down, accompanied by a gradual thinning of the oil film, to an arc of about 20 deg., at which the journal friction might reach its lowest value before bearing failure.

In such a high-speed bearing there would perhaps be a certain bearing-film arc at which the bearing temperature would remain constant or beyond which a shorter arc caused by a lower rate of supply might result in a cooler bearing. A point would eventually be reached, however, beyond which a further reduction of the oil-supply rate would result in partial metallic contact, higher friction, and then bearing failure.

In the three preceding paragraphs a high-speed bearing was studied. The same general reasoning would apply to a moderate-speed bearing, except that it would run cooler and cooler from the point where the film arc began falling below 360 deg. A minimum oil rate, however, would eventually be reached, below which the friction would increase and bearing failure soon occur.

The nature of the variation in bearing temperature and the minimum oil-supply rate below which bearing failure would occur, depend upon radiation, clearance, load, speed, oil viscosity, design, and, lastly but very important, the perfection of finish of the bearing surfaces, their materials, and the ability of the oil to resist seizure of the surfaces when the metallic contact begins.

Important conclusions may be drawn from the above discussion. A bearing may run quite cool and still have a very small margin of safety. An increase in its rate of oil supply will make it safer, although it may at the same time make it run hotter. The factor of safety actually provided depends, first, upon the knowledge and ideals of the designer, second, upon workmanship in the manufacture and installation, but lastly, upon proper care by user.

Lightly loaded bearings will run at fairly high surface speeds with a very deficient oil supply. Many bearings at moderate and low speeds run with partial metallic contact and wear, even with fairly high unit pressures, and do not burn out if a regular although deficient oil supply is provided. An oil supply may therefore be just sufficient to prevent failure but not to prevent wear, or it may

be increased to or beyond the point at which wear ceases, if the bearing is properly designed.

The charts, Figs. 36, 37, 38, and 46, are offered as only a step in the right direction, so that thoughtful designers may proceed more intelligently than heretofore. They must realize, but not overrate, the limitations to the methods employed by the author in getting his results.

Referring more in detail to the constructive criticism of this paper, the P_0 constant of integration mentioned by Mr. Karelitz was used in Eq. 3, Part I, for the full bearing where it appeared to serve a purpose. It does not appear in Eq. 20, Part III, because zero pressure is assumed to be whatever exists at each end of the partial-bearing film, usually atmospheric.

Professor Heck believes that at certain degrees of eccentricity, eddies or cross-flow occur and limit the range of application of Reynolds' equations. Although it is not easy to visualize the paths taken by oil particles within a wedge-shaped film, it appears to the author that accelerations and retardations of particles are not harder to understand at one eccentricity than at another. Reynolds fully realized that reverse flow might take place and was the first to represent them by diagrams. He therefore did not recognize any serious limitations to the applications of theory except variable viscosity and side leakage, which he pointed out.

Professor Heck has also shown what few before him have realized, namely, that Reynolds' fundamental equations can be derived much more easily than by the method Reynolds employed. It is hoped, therefore, that Professor Heck may find time to continue his study, and that he will find simple solutions of other elusive problems in lubrication, such as the quantitative effect of side leakage, and variable oil viscosity, in journal bearings.

Referring to Mr. Newbigin's comments, the author has not "assumed" but the two-dimensional theory "has shown" that a constant clearance ratio $\frac{\eta}{a}$ will produce the same pressure range in two bearings of different diameters, but of the same angular length and position, provided the oil viscosity is constant and the angular velocities of the journals are the same. Added to this, the same should be true, taking side leakage into account, if the ratio of bearing width to journal diameter is the same in both bearings.

Although Reynolds explained Tower's results analytically by assuming a running clearance, i.e., a difference in curvature between journal and bearing, the author believes he did so because it was the only simple assumption he could make and have the film complete from end to end of the bearing. If Tower's bearing had been made with a leading angle more than half the total bearing angle, there is little doubt but that Reynolds would have solved the problem by means of the simpler case of the fitted bearing. The author has found that there is nothing about the fitted bearing that

is inconsistent with its perfect operation. It is not more necessary to relieve the entering edge than it is to similarly relieve the edge of a fitted pivoted-thrust-bearing shoe.

Mr. Newbigin states that the use of pivoted pads in journal bearings becomes more advantageous as the bearing width is made shorter, but he does not state what the proportions are at which he would expect the single plain bearing and the two-pad pivoted bearing to be equally efficient when having the same width. Obviously a short bearing, whether pivoted or otherwise, must run with a higher unit pressure than a longer one on a given shaft. The higher the pressure, the better the workmanship must be to prevent seizure. The factor of safety must therefore not be overlooked. It is better to keep the bearing as simple as possible and improve its design and finish, rather than to make it more complicated and even have to resort to still greater care in design and finish, with doubtful advantage in the end. With full knowledge of the possibilities of simple journal bearings before him, the designer should have recourse to the pivoted type only when there is no better solution of his problem.

The roll-neck data given by Mr. Newbigin are admittedly not a fair comparison of the two types of bearings, because in one case there was the high friction of greasy surfaces and in the other the fluid friction of well-lubricated surfaces. The simple bearing, if properly designed, finished, and oiled, might have proved much superior to the pivoted pad bearing. It is therefore desirable that the two types of bearings be carefully compared by test under identical conditions, each type being designed, built, "groomed," and "ridden" in the trial by equally competent and careful engineers. The author would gladly welcome the receipt of the data on such tests.

A. G. M. Michell's three-dimensional study of the lubrication of plane surfaces, to which Mr. Newbigin refers, is a most creditable and valuable piece of work. It still has the limitation that it is based on the constant-viscosity assumption. Michell's findings, while pointing in the right direction, cannot be applied directly to journal bearings, because plain and journal bearings are so geometrically different.

So far as the author is aware, there is no *complete* mathematical study of lubricated surfaces. Hence it is desirable that factors, determined by careful test, be applied to such theoretical studies as have been made, so as to place them before designers in as usable a form as possible.

No. 1989

CHARTS FOR STUDYING THE OIL FILM IN BEARINGS

By George B. Karelitz, East Pittsburgh, Pa.[1]
Non-Member

It has been shown several times that the hydrodynamical theory of lubrication satisfactorily explains the mechanism of lubrication. Since the necessary calculations are very complicated, the practical application of the theory, up to the present time, has been very difficult. The charts presented in the paper give the designer or investigator a means of determining with sufficient accuracy the shape and pressures in the oil film for bearings under different conditions.

BEARINGS generally used at present for high-speed machines are of the central partial type, the arc of bearing varying in different constructions from 90 to 150 deg. A sufficient oil supply is provided for the building up of a "perfect" oil film, and for this case the mechanism of lubrication has been analyzed by several authors and shown to follow the hydrodynamical laws of flow of viscous fluids.

2 A cross-section of a bearing is shown in Fig. 1. The arc of bearing AB is central, so that $BF = FA$.

3 The general equation of the motion of oil in the case of a very long bearing surface where side leakage was neglected was given by N. Petroff [2] and O. Reynolds,[3] and is

$$\frac{dp}{dx} = -6\mu U \frac{h-h_1}{h^3} \quad \ldots\ldots\ldots [1]$$

where

p = oil pressure in the film
x is measured along the circumference of the shaft
μ = absolute viscosity of the oil
U = rubbing velocity of the shaft
h = variable thickness of the oil film
h_1 = thickness of the oil film in the point of maximum pressure.

[1] Westinghouse Research Laboratory.
[2] N. Petroff: New Theory of Friction. German edition by Wurzel, Hamburg, 1887.
[3] O. Reynolds: On Theory of Lubrication. Phil. Trans., Roy. Soc. of London, 1886.

Contributed by the Special Research Committee on Lubrication and presented at the Annual Meeting, New York, November 30 to December 4, 1925, of The American Society of Mechanical Engineers.

4 To form an approximate picture of the pressure distribution in an oil film of a bearing the following suppositions are made.

5 (*a*) The forces acting on the journal are the pressure of the oil acting normally to the shaft surface and the friction forces. Their resultant must be equal to the load on the bearing. The friction force being comparatively small, the load is taken as equal to the resultant of the forces due to oil pressure alone.

6 (*b*) The oil pressure in the annular clearance outside the carrying arc AB is taken as $p = 0$, as it is small (only a few lb. per sq. in. in a properly designed bearing) and may be neglected in comparison with the pressures existing in the carrying oil film. The vacuum which theoretically ought to be created under the bearing

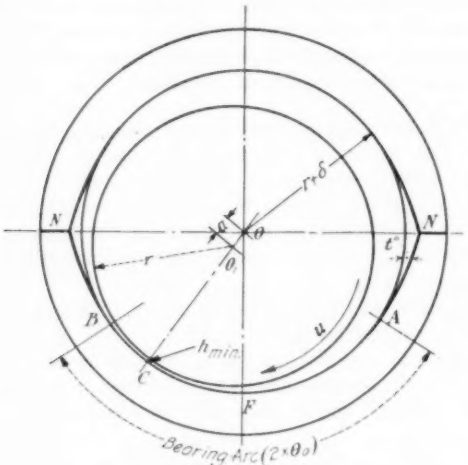


FIG. 1 CROSS-SECTION OF BEARING

cover cannot exist due to oil vapor being formed in the clearance. As to the wedge sections NB and NA, the angle t in Fig. 1 is large enough (above 2 deg.) to consider the pressure built up in the wedges as negligible. The pressure of the oil being $p = 0$ at A, grows to some maximum at a certain point with the corresponding thickness of film h_1 and falls down to the point C of nearest approach between journal and bearing. The pressure p at C is considered equal to zero (the pressure from C to B theoretically is negative).

7 (*c*) The experiments of Kingsbury, Lasche, and others have shown that the actual distribution of pressures in the oil film along the bearing axis follows very nearly a sine curve or a parabola of the second degree (Fig. 2). The assumption is made that in the middle plane of a bearing the pressure is the same as in a very long bearing (infinite length), the pressures in the oil uniformly de-

creasing to the ends according to one of the laws mentioned. It is evident that the carrying capacity of a bearing is only a part of what it would be with ends closed and pressures along the length uniformly equal to those in the middle plane. In the case of a sine-law distribution the carrying capacity is

$$s = 2/\pi = 0.637$$

and in the case of a parabolic-law distribution it is

$$s = \tfrac{2}{3} = 0.667$$

of what it would be with closed bearing ends. In the following

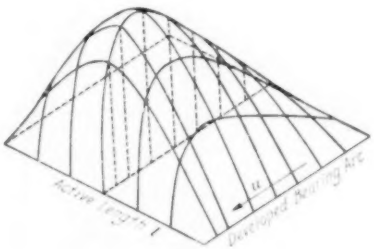


FIG. 2

calculations the value $s = \frac{2}{3}$ will be taken. The coefficient s may be called the longitudinal efficiency of the bearing.

8 (*d*) The bearing shell is considered as absolutely rigid, and the axis of the shaft as parallel to the axis of the bearing (no misalignment of shaft and bearing). It has to be borne in mind that the distortion of the bearing may affect considerably the distribution of pressures and the friction losses.

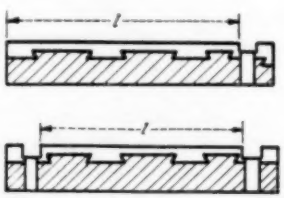


FIG. 3

9 (*e*) In the case of a bearing being furnished with an annular leak-off groove or two grooves at the ends, the pressure in the grooves is to be taken as $p = 0$, the active length of the bearing being the length between the groove and the opposite end of bearing or the length between grooves. See Fig. 3.

10 The basic equation [1] gives the pressures in the middle plane of a bearing. As

$$x = r\theta,$$

where r is the radius of the journal and θ is measured in radians,

$$dx = rd\theta$$

and

$$\frac{dp}{dx} = \frac{1}{r} \times \frac{dp}{d\theta}$$

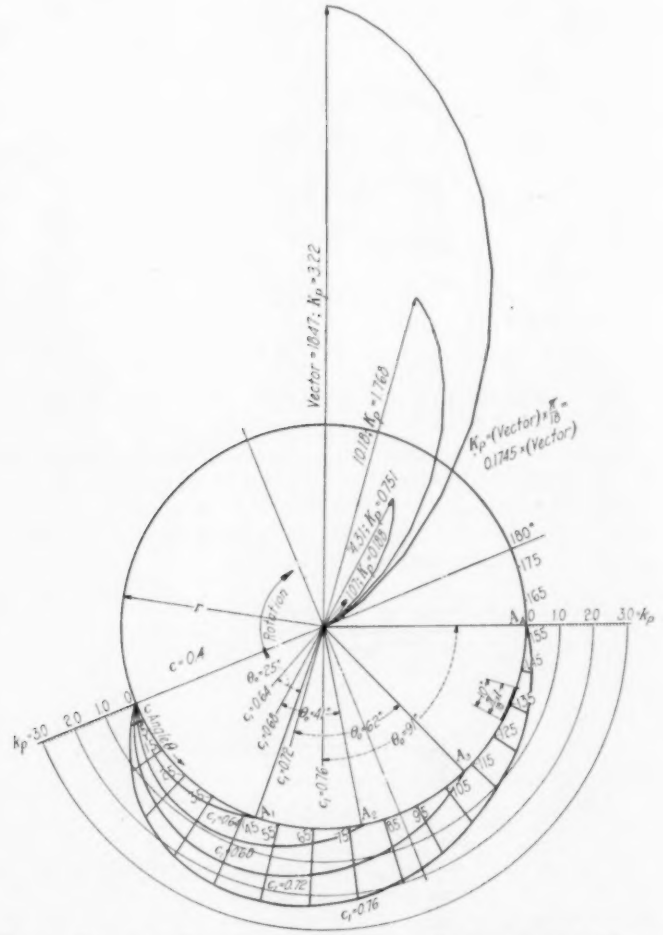


FIG. 4 CURVES OF PRESSURES EXISTING IN A BEARING FOR $c = 0.4$ AND VARIOUS VALUES OF c_1
(See Table 4.)

Taking $\theta = 0$ at the point C of nearest approach, the radius of the bearing being $r+\delta = r+mr = r(1+m)$, or the radial clearance being $\delta = mr$ (m is from 0.001 to 0.002), the eccentricity of the journal in the bearing being $a = c\times\delta = c\times m\times r$, with $0<c<1$, the variable clearance h is

$$h = \delta - a\cos\theta = \delta(1-c\cos\theta) = mr(1-c\cos\theta) \quad . \quad . \quad [2]$$

The smallest clearance is at $\theta = 0$. $h_{min} = \delta - a = r\times m(1-c)$ Substituting [2] in [1], we have

$$\frac{dp}{dx} = \frac{dp}{rd\theta} = -6\mu U\,\frac{mr(1-c\cos\theta)-h_1}{m^3r^3(1-c\cos\theta)^3} \quad . \quad . \quad . \quad [3]$$

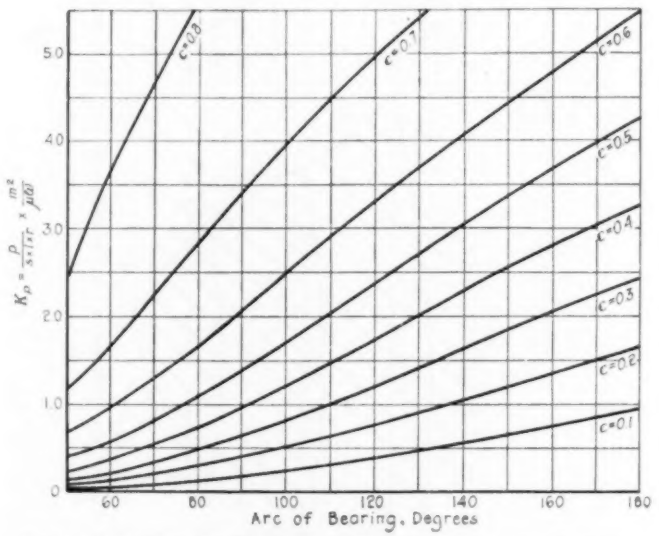


FIG. 5 RELATION BETWEEN ARC OF BEARING AND K_p FOR VARIOUS VALUES OF c

11 The value of h_1 is evidently larger than h_{min} and smaller than δ, or $(\delta-a)<h_1<\delta$; $rm(1-c)<h_1<mr$. Taking $h_1 = c_1mr$, where $(1-c)<c_1<1$, Equation [3] gives

$$\frac{dp}{rd\theta} = -6\mu U\,\frac{mr(1-c\cos\theta)-c_1mr}{m^3r^3(1-c\cos\theta)^3} = \frac{-6\mu U}{m^2r^2}\,\frac{(1-c\cos\theta)-c_1}{(1-c\cos\theta)^3},$$

or

$$\frac{dp}{d\theta} = -\frac{6\mu U}{m^2r}\,\frac{(1-c\cos\theta)-c_1}{(1-c\cos\theta)^3} = -\frac{6\mu\omega}{m^2}\,\frac{(1-c\cos\theta)-c_1}{(1-c\cos\theta)^3} \quad [4]$$

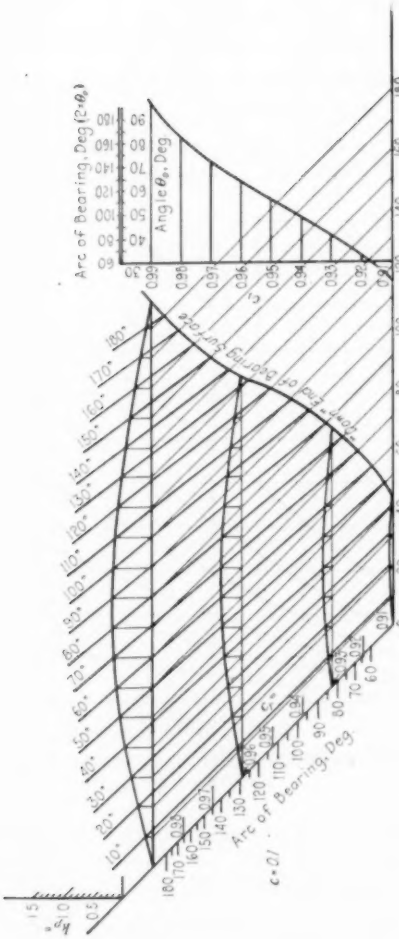


FIG. 6 CHART FOR DETERMINING PRESSURE OF OIL FILM FOR $c = 0.1$

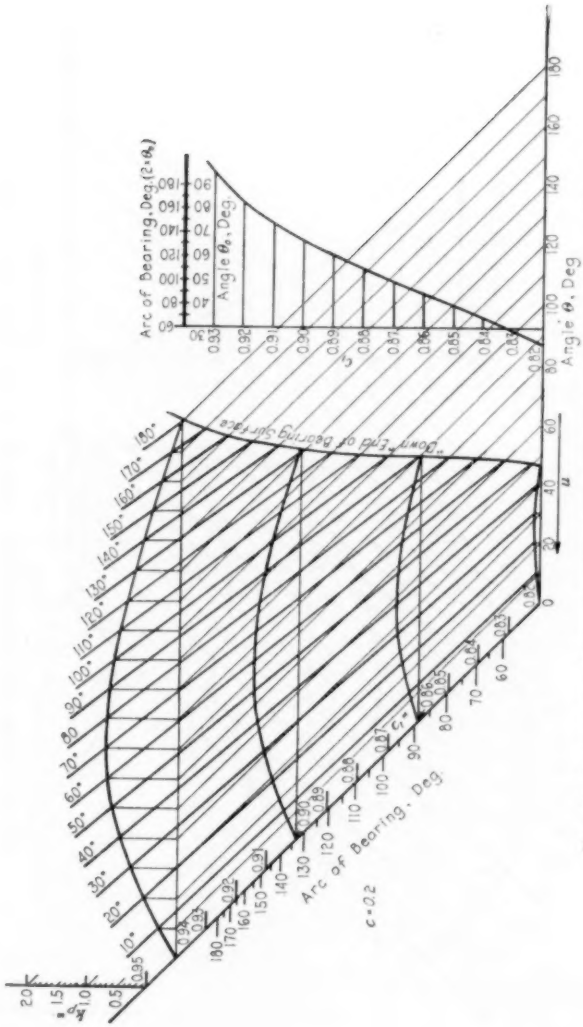


FIG. 7 CHART FOR DETERMINING PRESSURE OF OIL FILM FOR $c = 0.2$

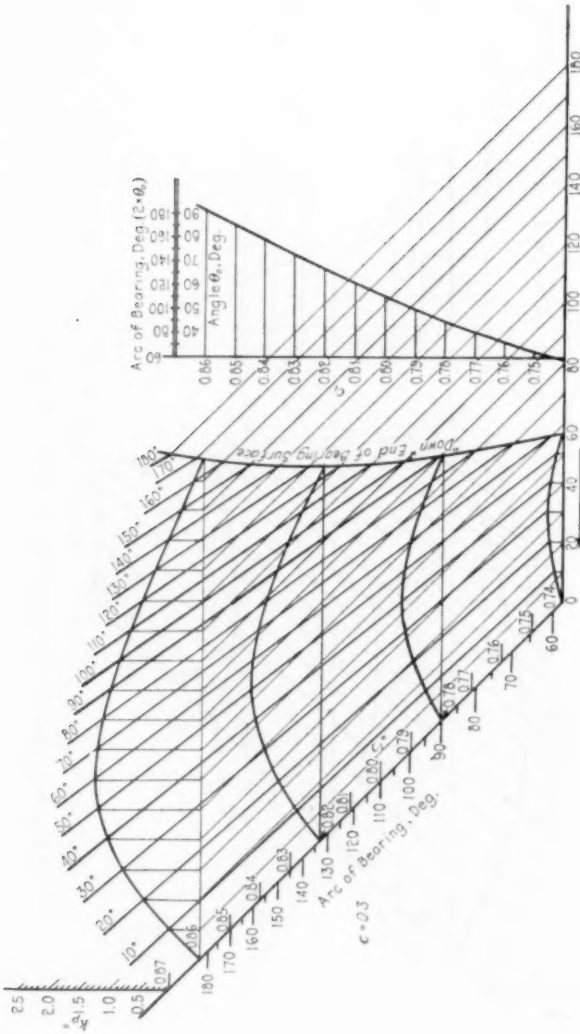


FIG. 8 CHART FOR DETERMINING PRESSURE OF OIL FILM FOR $c = 0.3$

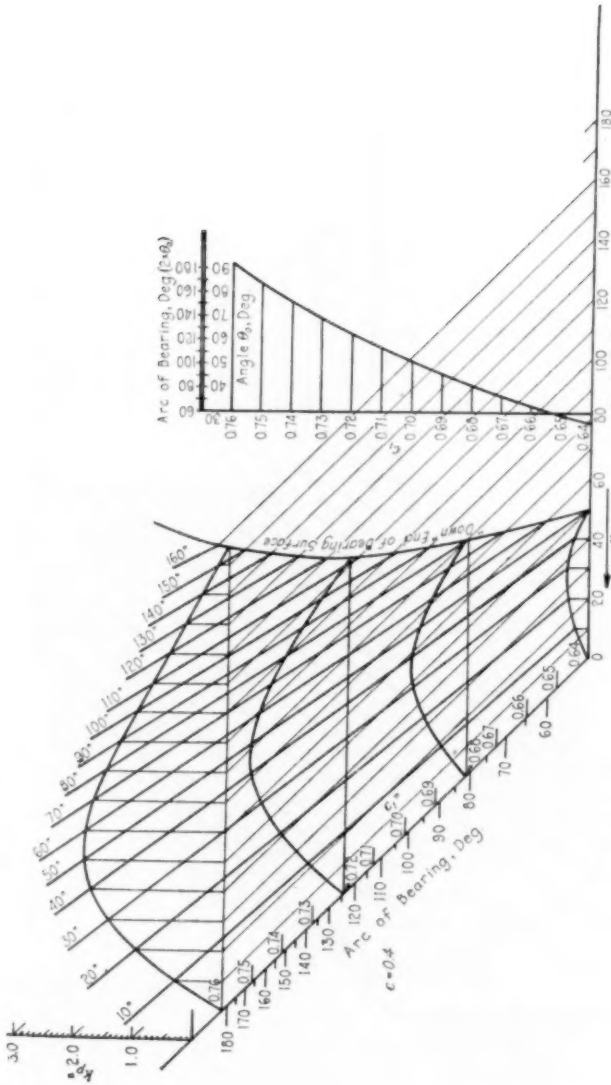


FIG. 9 CHART FOR DETERMINING PRESSURE OF OIL FILM FOR $c = 0.4$

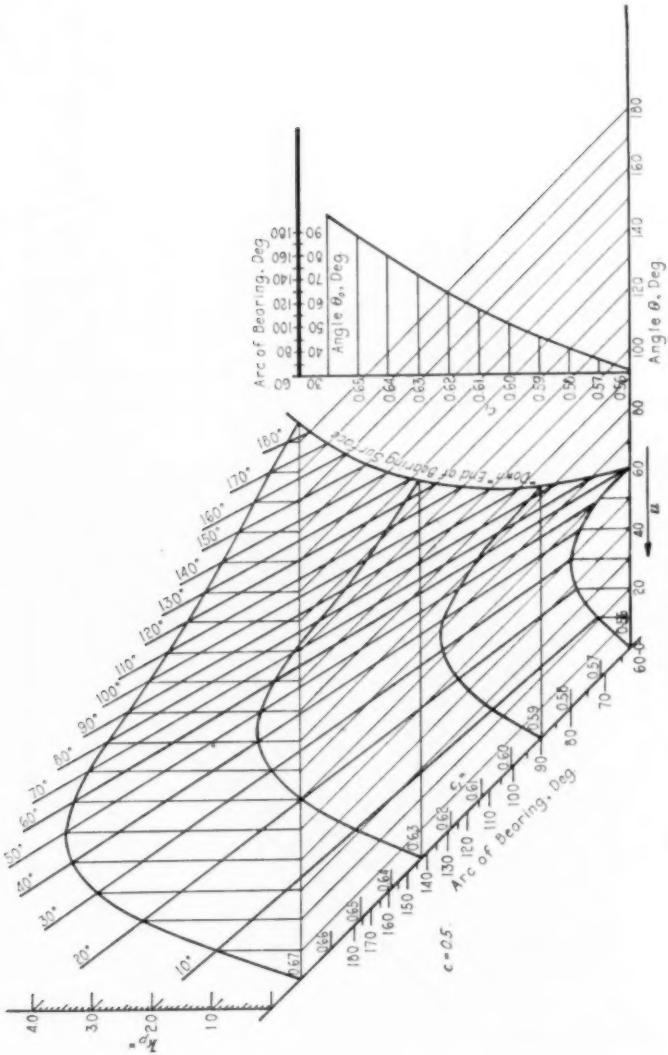


FIG. 10 CHART FOR DETERMINING PRESSURE OF OIL FILM FOR $c = 0.5$

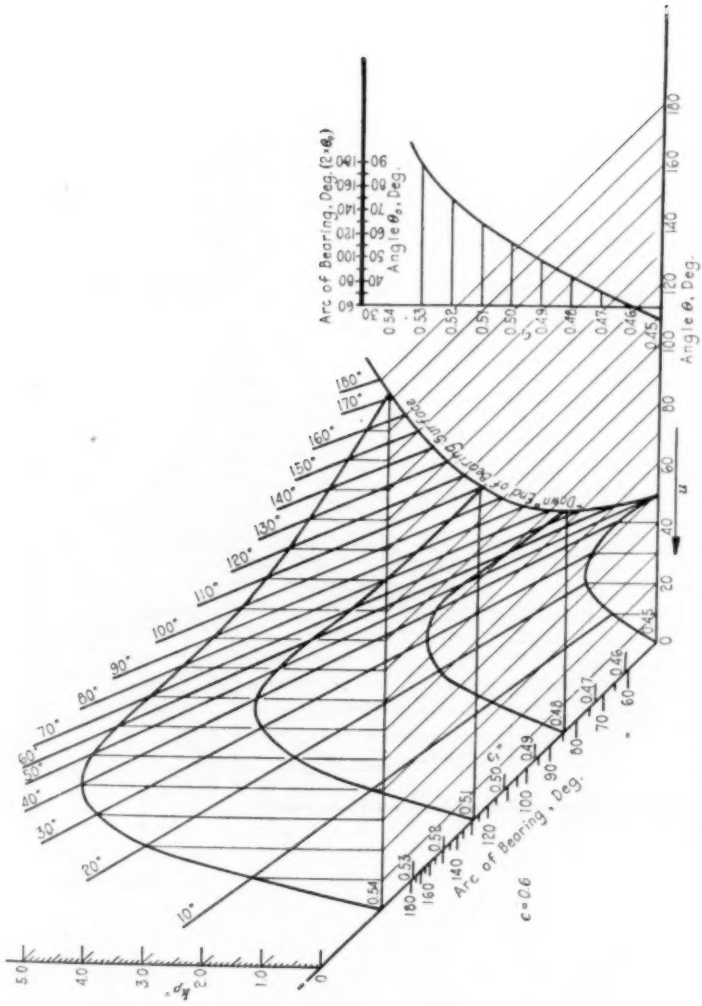


FIG. 11 CHART FOR DETERMINING PRESSURE OF OIL FILM FOR $c = 0.6$

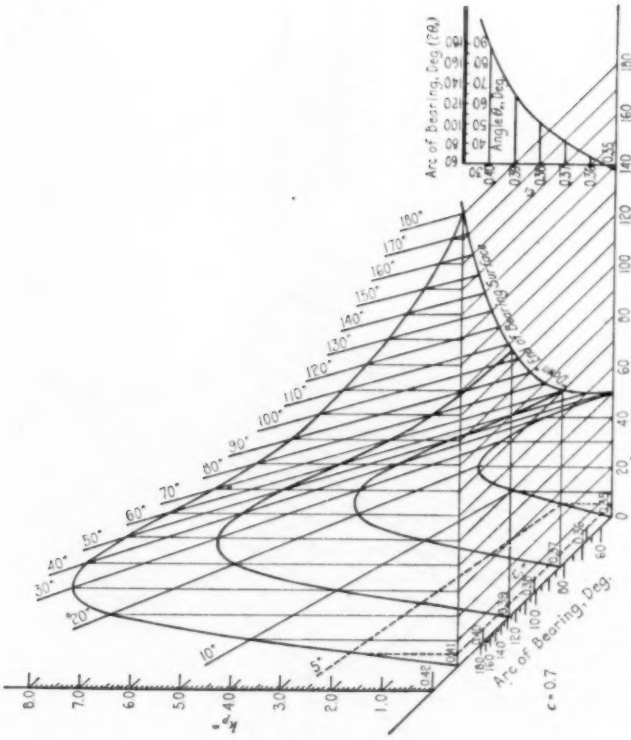


FIG. 12 CHART FOR DETERMINING PRESSURE OF OIL FILM FOR $c = 0.7$

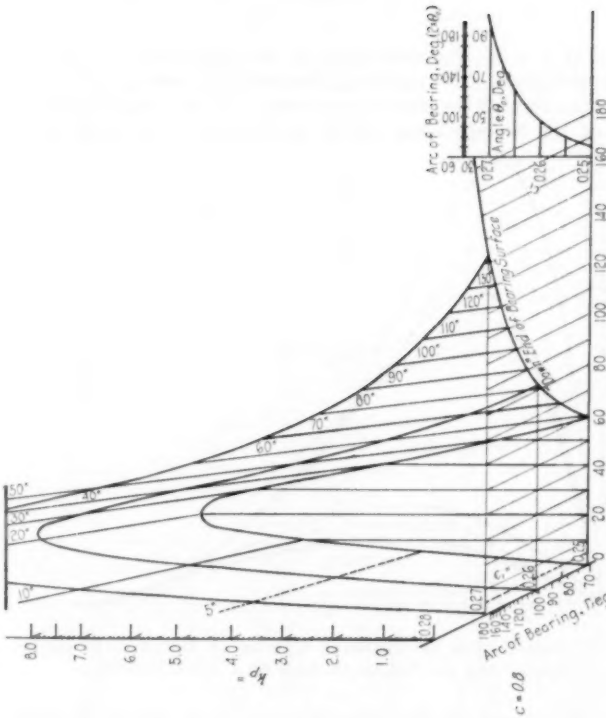


FIG. 13 CHART FOR DETERMINING PRESSURE OF OIL FILM FOR $e = 0.8$

where

$$\omega = U/r = \text{the angular velocity of the shaft.}$$

The exact solution of Equation [4] is

$$-\frac{m^2}{6\mu\omega}p = \frac{2c(1-c^2)-3cc_1}{2(1-c^2)^2}\frac{\sin\theta}{(1-c\cos\theta)} - \frac{cc_1}{2(1-c^2)}\frac{\sin\theta}{(1-c\cos\theta)^2} + \frac{(2-c_1)(1-c^2)-3c_1}{(1-c^2)^2\sqrt{1-c^2}}\tan^{-1}\left(\sqrt{\frac{1+c}{1-c}}\tan\theta/2\right)$$

if $p = 0$ at $\theta = 0$. The evaluation of the numerical values of p is too complicated, and graphical integration is more practical.

12 The values of the absolute viscosity μ, of the angular velocity ω, and of the bearing clearance m being given, the pressures in

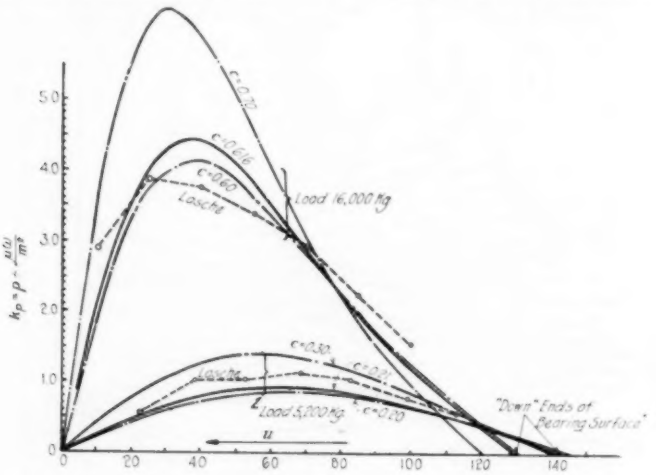


FIG. 14 COMPARISON OF PRESSURE CURVES AS DETERMINED FROM CHARTS AND AS FOUND IN LASCHE'S EXPERIMENTS

the oil film depend on the values of c and c_1, as shown by Equation [4]. Integrating,

$$p = -\frac{6\mu\omega}{m^2} \times \int \frac{(1-c\cos\theta)-c_1}{(1-c\cos\theta)^3}\,d\theta$$

$$= -\frac{6\mu\omega}{m^2}\left\{\int\frac{d\theta}{(1-c\cos\theta)^2} - c_1\int\frac{d\theta}{(1-c\cos\theta)^3}\right\} = k_p \times \frac{\mu\omega}{m^2} \quad [5]$$

where,

$$k_p = -6\left\{\int\frac{d\theta}{(1-c\cos\theta)^2} - c_1\int\frac{d\theta}{(1-c\cos\theta)^3}\right\} = p\times\frac{m^2}{\mu\omega}. \quad [6]$$

13 Tables 1 to 8 give the values of k_p for $c = 0.1, 0.2, \ldots . 0.8$ and several values of c_1. The integration was made by summation

of ordinates, the increment of θ being taken as 10 deg. Each value of c and c_1 represents a certain possible partial bearing, as shown for the case of $c = 0.4$ in Fig. 4. The point C is the point of nearest approach as in Fig. 1; the pressures corresponding to $c_1 = 0.64$; $c_1 = 0.68$; $c_1 = 0.72$ and $c_1 = 0.76$, as given in Table 4, are shown by curves CA_1, CA_2, CA_3 and CA_4 correspondingly.

TABLE 1 VALUES OF k_p FOR $c = 0.1$

$$k_p = p \div \frac{\mu\omega}{m^2}$$

θ deg.	$c_1 = 0.93$	$c_1 = 0.95$	$c_1 = 0.97$	$c_1 = 0.99$
5	0.022	0.038	0.059	0.065
15	0.064	0.107	0.150	0.193
25	0.098	0.170	0.241	0.312
35	0.119	0.219	0.317	0.416
45	0.128	0.252	0.378	0.503
55	0.121	0.271	0.422	0.573
65	0.097	0.272	0.447	0.622
75	0.056	0.253	0.452	0.650
85	−0.003	0.217	0.438	0.659
95		0.165	0.406	0.648
105		0.098	0.360	0.610
115		0.019	0.299	0.569
125		−0.071	0.227	0.526
135			0.145	0.460
145			0.055	0.387
155			−0.040	0.309
165				0.226
175				0.141
185				0.055
195				−0.030

TABLE 2 VALUES OF k_p FOR $c = 0.2$

$$k_p = p \div \frac{\mu\omega}{m^2}$$

θ deg.	$c_1 = 0.83$	$c_1 = 0.86$	$c_1 = 0.89$	$c_1 = 0.92$	$c_1 = 0.95$
5	0.031	0.062	0.093	0.124	0.155
15	0.086	0.177	0.268	0.359	0.451
25	0.121	0.271	0.422	0.572	0.722
35	0.127	0.332	0.537	0.742	0.948
45	0.097	0.354	0.612	0.869	1.127
55	0.031	0.336	0.642	0.947	1.252
65	−0.067	0.281	0.630	0.977	1.324
75		0.189	0.575	0.962	1.348
85		0.066	0.488	0.908	1.330
95		−0.081	0.373	0.825	1.278
105			0.236	0.717	1.200
115			0.083	0.590	1.097
125			−0.083	0.448	0.979
135				0.297	0.850
145				0.137	0.711
155				−0.026	0.567
165					0.418
175					0.267
185					0.114
195					0.064
205					0.012
215					−0.043

14 If a strip of bearing of one unit of length be taken, the pressure p is equal to the force applied to the journal surface per unit length of circumference. In Fig. 4 these forces are combined

TABLE 3 VALUES OF k_p FOR $c = 0.3$

$$k_p = p \div \frac{\mu\omega}{m^2}$$

θ deg.	$c_1 = 0.74$	$c_1 = 0.78$	$c_1 = 0.82$	$c_1 = 0.86$
5	0.061	0.122	0.183	0.250
15	0.167	0.347	0.528	0.709
25	0.229	0.524	0.817	1.112
35	0.229	0.627	1.023	1.420
45	0.161	0.650	1.139	1.629
55	0.028	0.597	1.166	1.733
65	− 0.160	0.476	1.114	1.750
75		0.307	1.005	1.696
85		0.101	0.845	1.587
95		− 0.130	0.656	1.441
105			0.446	1.267
115			0.225	1.078
125			0.003	0.877
135				0.673
145				0.463
155				0.256
165				0.046
175				− 0.163

TABLE 4 VALUES OF k_p FOR $c = 0.4$

$$k_p = p \div \frac{\mu\omega}{m^2}$$

θ deg.	$c_1 = 0.64$	$c_1 = 0.68$	$c_1 = 0.72$	$c_1 = 0.76$
5	0.098	0.194	0.292	0.389
15	0.258	0.543	0.829	1.114
25	0.327	0.785	1.243	1.701
35	0.277	0.884	1.493	2.106
45	0.108	0.842	1.576	2.310
55	− 0.155	0.681	1.517	2.354
65		0.436	1.354	2.273
75		0.138	1.121	2.107
85		− 0.187	0.848	1.882
95			0.554	1.631
105			0.256	1.368
115			− 0.042	1.098
125				0.831
135				0.569
145				0.313
155				0.062
165				− 0.188

TABLE 5 VALUES OF k_p FOR $c = 0.5$

$$k_p = p \div \frac{\mu\omega}{m^2}$$

θ deg.	$c_1 = 0.56$	$c_1 = 0.59$	$c_1 = 0.62$	$c_1 = 0.65$	$c_1 = 0.67$
5	0.252	0.377	0.503	0.628	0.713
15	0.747	1.115	1.481	1.848	2.091
25	0.959	1.537	2.112	2.689	3.073
35	0.918	1.670	2.416	3.166	3.667
45	0.664	1.549	2.430	3.314	3.902
55	0.266	1.252	2.234	3.218	3.874
65	− 0.205	0.857	1.913	2.969	3.675
75		0.415	1.530	2.642	3.383
85		− 0.030	1.127	2.278	3.049
95			0.727	1.914	2.703
105			0.347	1.559	2.364
115			− 0.013	1.220	2.042
125				0.897	1.730
135				0.592	1.433
145				0.302	1.151
155				0.023	0.878
165				− 0.247	0.616
175					0.355
185					0.099
195					− 0.155

vectorially and the resultants are found. The carrying capacity of the bearing per unit length is then

$$K_p \times r \times \frac{\mu\omega}{m^2} \quad \ldots\ldots\ldots\ldots [7]$$

where K_p is the load characteristic of the bearing strip and is found as shown in Fig. 4. The total load on the bearing being P, the active length of the bearing being l, and the longitudinal efficiency of the bearing being s, the carrying capacity of the middle element of it has to be

$$1/s \times P/l = P/sl$$

and

$$K_p \times r \times \frac{\mu\omega}{m^2} = P/sl$$

or

$$P = K_p \times \frac{r\mu\omega}{m^2} \times s \times l \quad \ldots\ldots\ldots [8]$$

or

$$K_p = \frac{P}{slr} \times \frac{m^2}{\mu\omega}$$

TABLE 6 VALUES OF k_p FOR $c = 0.6$

$$k_p = p \div \frac{\mu\omega}{m^2}$$

θ deg.	$c_1 = 0.46$	$c_1 = 0.48$	$c_1 = 0.51$	$c_1 = 0.54$
5	0.491	0.655	0.901	1.146
15	1.269	1.740	2.443	3.147
25	1.571	2.292	3.374	4.457
35	1.377	2.287	3.652	5.018
45	0.844	1.888	3.451	5.016
55	0.147	1.281	2.981	4.682
65	− 0.586	0.610	2.402	4.193
75		− 0.047	1.809	3.661
85			1.247	3.144
95			0.733	2.662
105			0.270	2.224
115			− 0.145	1.827
125				1.463
135				1.132
145				0.823
155				0.531
165				0.249
175				− 0.025

The directions of the resultant vectors give the angles θ_0. The points A_1, A_2, A_3 represent the corresponding " down " ends of the active bearing surface, and the bearing being a central partial one, the resultant force has to be collinear with the bearing load, i.e., in the central plane; hence the angles θ_0 (Fig. 4) are equal to one-half of the bearing arc AB. It is evident that two fixed values of c and c_1 represent a certain possible central partial bearing of definite carrying capacity K_p and bearing arc $2 \times \theta_0$.

15 Diagrams similar to Fig. 4 were drawn for each value of $c = 0.1$ to $c = 0.8$, and subsequent graphical interpolation has

given the charts Fig. 5 to Fig. 13 for studying the oil film in a bearing.

16 Chart No. 1 (Fig. 5) gives the value of c corresponding to the given bearing arc and load characteristic K_p. The smallest thickness of the oil film is then found as

$$h_{\min} = mr(1-c)$$

TABLE 7 VALUES OF k_p FOR $c = 0.7$

$$k_p = p \div \frac{\mu\omega}{m^2}$$

θ deg.	$c_1 = 0.357$	$c_1 = 0.377$	$c_1 = 0.397$	$c_1 = 0.417$
5	1.103	1.491	1.880	2.268
15	2.727	3.814	4.901	5.988
25	3.114	4.723	6.332	7.941
35	2.482	4.435	6.387	8.339
45	1.363	3.524	5.686	7.847
55	0.148	2.436	4.724	7.012
65	− 0.968	1.394	3.760	6.124
75		0.482	2.895	5.306
85		− 0.291	2.151	4.593
95			1.519	3.983
105			0.981	3.459
115			0.518	3.008
125			0.113	2.611
135			− 0.249	2.255
145				1.931
155				1.631
165				1.347
175				1.068
185				0.794
195				0.515
205				0.250
215				− 0.010

TABLE 8 VALUES OF k_p FOR $c = 0.8$

$$k_p = p \div \frac{\mu\omega}{m^2}$$

θ deg.	$c_1 = 0.25$	$c_1 = 0.26$	$c_1 = 0.27$	$c_1 = 0.275$
5	3.272	3.926	4.581	4.907
15	7.397	9.147	10.896	11.771
25	7.394	9.824	12.253	13.468
35	5.230	8.017	10.804	12.198
45	2.859	5.721	8.688	10.072
55	0.601	3.659	6.718	8.247
65	− 1.096	2.010	5.117	6.671
75		0.736	3.871	5.438
85		− 0.249	2.901	4.477
95			2.137	3.718
105			1.521	3.105
115			1.013	2.600
125			0.582	2.170
135			0.207	1.797
145			− 0.128	1.463
155				1.156
165				0.870
175				0.593
185				0.319
195				0.048
205				− 0.231

17 It has to be noted that the value μ to be used for determination of K_p is the viscosity of oil in the oil film proper. It may be found approximately by taking into consideration that the average temperature of the oil film is from 5 to 10 deg. cent. higher than the temperature of the outflowing oil. The value of

the viscosity is to be converted into units corresponding to the rest of the data, the conversion coefficients are

$$1 \text{ poise} = 1.02 \times 10^{-6} \text{ kg-sec. per sq. cm.}$$
$$= 1.45 \times 10^{-5} \text{ lb-sec. per sq. in.}$$

Using the chart corresponding to the determined value of c or interpolating between two corresponding charts the pressure curve for the oil film is plotted for the given bearing arc. The point of nearest approach is given by the length of the active oil film.

18 The use of the charts is seen from the following example. O. Lasche in his experiments[1] of 1918 used a bearing with the following elements:

$$\text{Diameter} = d = 20 \text{ cm.} = 7.87 \text{ in.}$$
$$\text{Radius} = r = 10 \text{ cm.} = 3.94 \text{ in.}$$
$$\text{Active length} = l = 40 \text{ cm.} = 15.75 \text{ in.}$$
$$\text{Clearance} = 0.34 \text{ mm. or } m = 0.0017 \text{ in.}$$

The rubbing speed at one series of tests was 30 meters per sec. or

$$U = 3000 \text{ cm. per sec.} = 98.4 \text{ ft. per sec.}$$
$$\text{Angular velocity} = 300 \text{ radians per sec.}$$
$$\text{Loads on bearing} = P = \begin{cases} a & 5200 \text{ kg.} = 11460 \text{ lb. (92.3 lb. per sq. in.)} \\ b & 16000 \text{ kg.} = 36300 \text{ lb. (284 lb. per sq. in.)} \end{cases}$$

The bearing arc was 150 deg. No data on the oil are given. Assuming the oil to be equivalent to "heavy turbine oil" the viscosities were approximately:

Case	Temperatures	Assumed viscosity
a	57 deg. cent.	0.146 poise $= 0.149 \times 10^{-6}$ kg-sec. per sq. cm.
b	66 deg. cent.	0.121 poise $= 0.123 \times 10^{-6}$ kg-sec. per sq. cm.

19 Taking the longitudinal efficiency as $s = \frac{3}{4}$, the load characteristic K_p (Equation [8]) is in case a

$$K_p = 1.26$$

From chart No. 1 for a bearing arc of 150 deg. and $K_p = 1.26$, c is found as $c = 0.21$. From charts Nos. 3 and 4 ($c = 0.2$ and $c = 0.3$) the pressure curves are plotted for the bearing arc of 150 deg. and by interpolation the curve $c = 0.21$ is found. In case b the load characteristic is

$$K_p = 4.66$$

20 From chart No. 1 for bearing arc of 150 deg. and $K_p = 4.66$ the value of c is found $c = 0.615$. From charts Nos. 7 and 8 ($c = 0.6$ and $c = 0.7$) the pressure curves for the bearing arc of 150 deg. are plotted (Fig. 14) and the curve for $c = 0.615$ is found by interpolation.

[1] O. Lasche: Konstruktion und Material im Bau von Dampfturbinen und Turbodynamos, Berlin, 1920.

21 O. Lasche gives the observed pressure curves for the Cases *a* and *b*. The pressures were converted into corresponding values of k_p by Equation [6] and plotted for comparison in Fig. 14. The coincidence of the curves can be considered as sufficient to justify the use of the charts for analyzing the behavior of a given bearing as to oil pressures and oil film thickness. It can also be seen that the longitudinal efficiency is somewhat higher than $\frac{3}{4}$ in Case *b* and lower in Case *a*. This is to be expected since the oil film thickness in Case *b* ($h_{min} = 0.0025$ in.) is considerably less than in Case *a* ($h_{min} = 0.0054$ in.).

DISCUSSION

H. A. S. HOWARTH.[1] The author has assumed that the pressure curve along the middle plane of the bearing — in the direction of rotation — is identical with that given by Reynolds and Harrison for a given shape and relative thickness of oil film. He has then assumed that, if side leakage is taken into account, the pressure variation toward the sides of the bearing would follow the parabolic law. By this means he figures that the carrying capacity of a given film will be $\frac{3}{4}$ of what it would be if side leakage were neglected.

Side leakage cannot be taken account of so easily because side leakage prevents the pressures at the middle plane from rising as high as they otherwise would. This is to be expected and it has been mathematically covered by A. G. M. Michell for plane bearing surfaces. Another feature to be remembered is that the law of variation and magnitude of the pressure at the middle plane changes with the ratio of bearing width to arcuate length.

Therefore if the author's factor ($s = \frac{3}{4}$) is approximately correct for a relatively wide bearing, it would be much too large for a narrow bearing. Hence he should give a table of values of *s* showing how it would vary with the ratio of bearing width to arcuate length. Such a table might be obtained by a prodigious amount of mathematical work. It would be better, however, to determine the values experimentally, as suggested by the writer in Part II of his Graphical Study of Journal Lubrication,[2] because experimentally determined factors would take variable viscosity as well as side leakage into account.

M. D. HERSEY.[3] In his comparison with Lasche's experiments in which the viscosity of the oil was not reported, the author assumes that Lasche used a medium machine oil; but if something quite different was used, in what way and to what extent would it affect the results of the present paper?

[1] Chief Engineer, Kingsbury Machine Works, Philadelphia, Pa. Mem. A.S.M.E.

[2] Trans. A.S.M.E., vol. 46 (1924), p. 832.

[3] Physicist, U. S. Bureau of Mines, Pittsburgh, Pa. Mem. A.S.M.E.

J. M. LESSELLS.[1] It is generally known that the work which Prof. H. F. Moore[2] did on the breakdown pressure of an oil film established the relation between P, the critical film pressure, and V, the speed of the journal. He discussed the correlation between P, the critical load which the oil film will stand, and V, the velocity. Last year another paper was presented dealing with experimental work[3] in which it was found that these critical bearing pressures were very much in excess of what they were shown to be by Moore. Possibly the author's work has shown that the oil film can withstand these enormous pressures and that the breakdown point cannot be due to excessive pressure but must arise from other causes, such as conditions of oil flow or state of machinery.

CHESTER B. HAMILTON.[4] There is one point that does not seem to be covered either in the present paper or in Mr. Howarth's paper.[5] That point is the difference of temperature between the shaft and the journal shell which will cut down the average thickness of the oil film. The shaft is liable to dissipate less heat than the metal of the bearing shell. It will therefore expand and reduce the thickness of the film. Such a fact might explain difficulties in the fitted bearings, because the radius of curvature of the exterior bearing in those cases might not be equal to, but be less than, the radius of curvature of the shaft.

THE AUTHOR. In reply to Mr. Hamilton, the question of trouble due to shaft expansion can be solved only by actual experience. In general, it is doubtful if this ever occurs in partial bearings because: (1) The temperatures of the shaft and bearing shell do not differ very much, the difference being only several degrees, even in the case when the shell is pressed into its housing giving a very good dissipation of heat; (2) even with a fitted bearing as soon as the journal is started an oil film is built up between bearing and journal, provided the upper half of the bearing is machined to allow motion of the journal. Consequently there is little probability of seizure occurring due to thermal change of radii of bearing and journal. Excessive distortion of the cylindrical shape of bearing shells due to change of temperature is sometimes responsible, however, for bearing trouble.

[1] Engineer, Research Dept., Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa. Mem. A.S.M.E.

[2] *American Machinist*, Aug. 1903.

[3] An Investigation of the Critical Bearing Pressures Causing Rupture in Lubricating-Oil Films, by L. N. Linsley. Trans. A.S.M.E., vol. 46 (1924), p. 855.

[4] President and Mechanical Engineer, Hamilton Gear & Machine Co., Toronto, Ontario, Canada. Mem. A.S.M.E.

[5] A Graphical Study of Journal Lubrication (Part III). See page 1073.

As regards Mr. Hersey's remarks, the example of Lasche was taken only as an illustration. It is a pity we have no data on the oil he used, since all other results are carefully recorded. The oil used by Lasche could not be very different, however, from the one assumed in the paper. An increase in the viscosity of the oil would result in a proportional decrease of k_p and consequent decrease of eccentricity and flattening out of the pressure curve; decrease in viscosity would give a sharper peak on the curve. Of course, as mentioned in the paper, the longitudinal efficiency might be slightly different from $\frac{2}{3}$.

The author agrees with Mr. Howarth that a table for values of the coefficient s is necessary. Experiments, such as made by Kingsbury and Lasche, have shown that for ratios of diameter to length as used at present, the longitudinal distribution of pressures justifies the author's assumption. For long bearings or those having small clearances, s will probably be near 0.70, and for short bearings and large clearances, near 0.60.

It has to be borne in mind that the paper gives only approximate solutions, and refinements can be considered only after a long series of experiments. Since, however, such experiments are difficult to make due to their high cost, we have to be satisfied with approximate solutions.

The question of film rupture raised by Mr. Lessells is very important. Experiments by Kingsbury and by Stanton (National Physical Laboratory, England) have proved quite convincingly that an oil film cannot be ruptured by any pressure met with in actual bearings. The rupture is due to mechanical contact of journal and bearing which is caused by imperfections of the rubbing surfaces piercing the film, or by misalignment of journal and bearing, or by distortion of the bearing shell. The thickness of the oil film when the contact occurs has actually been measured by Vieweg (Physikalisch-Technische Reichsanstalt, Germany).

The question of critical pressure therefore becomes, in a last analysis, one of minimum allowable thickness of oil film. A committee should be appointed to determine the allowable film thickness and hence the maximum load a bearing may carry. This limit should also take into account the condition of probable allowable misalignment, as well as the roughness due to commercial finish of the rubbing surfaces.

No. 1990

RADIATION IN BOILER FURNACES

By B. N. Broido,[1] New York, N. Y.

Member of the Society

There is now a noticeable tendency toward an extensive application of water-cooled furnace walls for absorbing heat by radiation. An analysis of the fundamentals of radiation and of the effect of water-cooled walls on the gas temperatures in the furnace is given in this paper.

There is a certain relation between the amount of fuel burned, the surface exposed to direct radiation in the furnace, and the total heat absorbed by this surface. Curves are given showing this relation for different installations and a standard curve is drawn which enables the designer to determine with sufficient accuracy for practical purposes what part of the total heat generated in the furnace at different ratings is absorbed by the water-cooled walls. This curve can also be used to find the heat transmission by radiation per square foot of heating surface for any given condition of the furnace.

The amount of radiant-heat-absorbing surface that can be installed advantageously in a furnace depends upon the type of furnace and the kind of fuel burned. Based on a number of installations with water-cooled walls, the most advantageous arrangement of radiant-heat-absorbing surface for given conditions is suggested.

The effect of radiation from gases on heat transmission, particularly with high temperature differences, and the influence of radiation upon the measurement of gas temperature, are briefly discussed.

THE rapid development of the steam turbine and the possibility of concentrating many power generators in large power stations have created a demand for steam in increased quantities. The logical progress has been, therefore, toward larger furnaces, larger boilers, and higher evaporation per square foot of boiler heating surface. With larger boilers and correspondingly larger furnaces, the maintenance of furnace walls becomes quite a problem, as the firebrick wall will not stand the high temperatures which occur when a boiler is operated at high efficiency. It would

[1] Consulting Engineer, The Superheater Company.

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be necessary to admit excess air through the fire to reduce the furnace temperature to a degree that the walls would stand. This difficulty has been overcome, at least partially, by using hollow air-cooled bricks and, more recently, by using water-cooled walls.

2 Attempts have been made previously to water-cool the furnace walls of water-tube boilers, but no serious consideration was given to them until some of the boilers in the Hell Gate Station of the United Electric Light and Power Company of New York were equipped with so-called fin walls. These installations proved conclusively that not only was the maintenance of the furnace decreased but the capacity of the boilers was appreciably increased. The water-cooled side walls of the Hell Gate boilers not only replaced practically all the brickwork in the side walls, but afforded considerable protection to the front and rear walls by absorbing the heat which they reflected and lowering the temperature of the furnace. In view of these results there is now a noticeable tendency toward a more extensive application of water-cooled walls.

3 This development has caused more thought to be given to the law of heat radiation than ever before. Tubes placed in a furnace absorb heat from the gases before they reach the boiler tubes, and this reduces the temperature in the furnace. The water-cooled walls generate a considerable amount of steam, making it possible to reduce the heating surface of the boiler proper for the same total evaporation. With a moderate amount of cooling surface in the furnace, it is possible to burn the fuel with a higher percentage of CO_2 and a correspondingly higher efficiency without the ill effect on the brickwork which is usually encountered with CO_2 contents and high furnace temperatures when water-cooled walls are not used. If the temperature of the furnace is lowered considerably, difficulty may be experienced in getting complete combustion with some kinds of fuel. The temperature of the gases in the furnace or passing through the boiler has considerable effect upon the superheater, which depends upon the heat absorbed from the furnace gases.

4 It is evident, therefore, that all designers of furnaces, boilers, and superheaters should know (1) what quantity of water can be evaporated by a given area of radiant-heat-absorbing surface, (2) what will be the influence of this surface on furnace temperature and combustion of the fuel, and (3) what amount of heating surface can be placed in a furnace to give the best results under given conditions. A discussion of these problems is the object of this paper.

Radiation and the Radiation Coefficient

5 Of the three well-known methods by which heat may be transmitted, viz., by radiation, by conduction, and by convection, the last two occur within a body or a combination of bodies in such

a way that the heat of these bodies is transmitted from one point to another by means of the particles of the body or through the body itself.

6 Heat transmission by radiation, however, occurs only from surface to surface. The heat travels from one body, or from one particle of the body, to another without giving up any heat to the air, gas, or any other matter that may fill the space between the two bodies or two particles of the body. This method of heat transmission is therefore the most difficult to understand and can be identified with the theory of the vibration of the ether. Physically, heat and light radiation are closely related. In accordance with our accepted ideas, both are electromagnetic waves, and differ from each other only by their wave lengths.

7 It is comparatively easy to prove that heat radiation has no effect upon the air between the body which radiates heat and the one which absorbs it. If one stands at some distance from a furnace provided with an easily opened door, when this door is quickly opened he feels the heat at practically the same instant he sees the fire, although the air between cannot by any means become so quickly heated. This shows that the heat passes through the air without heating it.

8 It is extremely difficult to determine by test the amount of heat radiated or absorbed. The difficulty exists mainly because it is very hard to separate the heat absorbed by radiation from that absorbed by convection, as these usually take place together. For this reason tests made by scientists have given different results, and the coefficients determined vary widely. Numerous formulas have been established for determining the heat absorbed by radiation. Not only do the coefficients vary widely, but also the relation between the heat transmitted by radiation and the temperatures is differently expressed.

9 For the last few decades, however, the practical engineering world has accepted the formula established by Stefan in accordance with which the radiation emanating from a body is proportional to the fourth power of the absolute temperature. Boltzmann later proved this formula theoretically correct for the case of black-body radiation. His proof was based on the electromagnetic theory of light radiation, although some experimenters have found considerable deviation in the case of bodies not conforming to black-body conditions. While there are no absolute black bodies, fortunately, however, the inside walls of a completely enclosed space have the same effect as a black body, and this, to a certain extent at least, applies also to a boiler furnace. Also any sooted surface or the surface of a bed of burning coal radiates and absorbs heat practically as a black body.

10 For practical purposes the Stefan-Boltzmann formula is sufficiently correct for non-transparent bodies. It includes a coefficient which depends upon the material and the condition of the

surface. It is questionable, however, whether this formula applies also to radiation from gases. It may be of interest to know how the coefficients for different materials were found.

11 The standard Stefan-Boltzmann formula for radiation is

$$Q = Cf\left[\left(\frac{T_1}{1000}\right)^4 - \left(\frac{T_2}{1000}\right)^4\right]$$

It shows that if a small surface f at the absolute temperature T_1 is completely surrounded by another surface at an absolute temperature T_2 so that all the heat radiated from f is absorbed by this surface, the amount of heat in B.t.u. that will be absorbed per sq. ft. per hr. will be Q, provided that the radiation coefficient is C. The coefficient of radiation C depends entirely upon the materials used and the condition of both surfaces, and is

$$C = \frac{1}{\frac{1}{c_1} + \frac{1}{c_2} - \frac{1}{c}}$$

c_1 and c_2 are respectively the radiation coefficients of the surfaces and c is the radiation coefficient of a black body, which has been established by tests to be 1600. It may be readily comprehended that if both surfaces are of the same condition or have the same coefficient, then c_1 is equal to c_2 and

$$C = \frac{1}{\frac{2}{c_1} - \frac{1}{c}} = \frac{1}{\frac{2}{c_1} - \frac{1}{1600}}$$

12 Tests were made with plates of the same material and under the same conditions, but with different temperatures, and with heat radiated from the plate at the higher temperature to the plate at the lower temperature. In this way the coefficient of radiation was found to be as follows:

For candle soot	1543
For unpolished brass	357
For roughly polished copper	276
For oxidized steel	1530
For highly polished steel	462
For rough or highly oxidized cast iron	1557
For brick	1494

13 Among the above, the coefficients of candle soot, oxidized steel, and brick are particularly important in connection with radiation in boiler furnaces.

14 With the use of the given coefficients, the Stefan-Boltzmann formula can be used in calculating radiant heat under different conditions. It is, however, correct only with the provision that the radiant surface is completely surrounded by the heat-absorbing surfaces. The formula gives the entire amount of heat radiated

by the surface f. As a matter of fact, the formula is correct only when f is a point. In practice this is seldom the case.

15 The principal problem is to determine the amount of heat which is radiated from one surface to another surface of a given size and at a certain position relative to the first.

16 In this case the general form of the Stefan-Boltzmann formula is

$$Q = CF\phi \left[\left(\frac{T_1}{1000} \right)^4 - \left(\frac{T_2}{1000} \right)^4 \right]$$

where ϕ represents the distances and relative positions of the surfaces.

17 Even in a case where the radiating surface is entirely surrounded by the absorbing one, the distance between any two points of the two surfaces being different, the relation between the surfaces or the solid angle must be considered and ϕ must be determined for the given conditions. In boiler furnaces the radiant as well as the absorbing areas have one or more flat or curved surfaces which involve very complicated calculations in determining ϕ.

18 In furnaces which consist of brick walls, further complications are introduced, as some of the radiant heat of the flame which strikes against the brick walls is absorbed by the wall and reradiated to the cold surfaces of the boiler, while the remainder is reflected from the brick back to the grate or directly to the cold surface. In this case determination of radiant heat actually utilized for the generation of steam requires a multiplicity of mathematical calculations.

19 The method for computing the radiant heat that can be absorbed in boilers with enclosed furnaces, as, for instance, in Scotch marine boilers, vertical fire-tube or Manning boilers, and in the firebox of a locomotive boiler, can be found in the work of M. Gerbel entitled Die Grundgesetze der Waermestrahlung. Roszak and Veron have developed very elaborate calculations for determining the radiant heat in water-tube boilers fired by coal on grates in an article entitled Le rayonnement calorifique envisagé du point de vue des application industrielles, published in the August, September, and October issues of the *Revue de Metallurgie*, 1924.

20 Translations of the above two treatises have been placed by the author in the files of the Society in connection with this paper, and are available to any one desiring to make an analytical study of the subject.

21 The excellent paper [1] by Wohlenberg and Morrow has indeed contributed to engineering literature by elucidating an analytical method for the calculation of radiation within a boiler furnace fired with pulverized fuel.

[1] Radiation in the Pulverized-Fuel Furnace. See p. 127 ante.

22 Thus, these three discourses taken together present a complete theoretical analysis of radiation in connection with steam boilers, as each deals with one of the three main types of boiler furnaces. They may therefore be of great assistance for a comprehensive analysis of the radiation effect in a furnace. For the designing engineer who lacks sufficient time for complicated calculations, these works are chiefly of academic value; and he requires data worked out in accordance with tests and theoretical analysis to determine the heat absorbed by radiation with accuracy sufficient for practical purposes.

23 The first two of the above references are based on the assumption that the heat is radiated from the fuel bed, but do not take into consideration the radiation from the flame or the gases. Wohlenberg and Morrow have dealt with radiation from the smaller suspended particles of coal by assuming the radiation of the flame to be of minor importance. It is the opinion of the author that more extensive consideration must be given to the flame and gases in radiation in a furnace.

24 It is interesting to note that radiation of flame has been the subject of many experiments by a number of scientists, and it is peculiar that the results differ so widely. A brief résumé of some of these results and experiments can be found in an article, Radiation from Non-Luminous Flames, by R. T. Haslam, W. G. Lovell, and R. D. Hunnerman, in *Industrial and Engineering Chemistry*, March, 1925.

25 Of particular interest is the test by Callendar and Julius. They have shown that the radiation from a Bunsen flame is almost entirely due to the CO_2 and steam vapor; also that the radiation depends upon the CO_2 or H_2O molecules formed without regard to the nature of the fuel used for combustion. The radiation from these flames was measured by Callendar by comparison with radiation of a black body, using the same pyrometer for both. It was found that with six flames having a total thickness of 8½ in., the radiation was the same as that of a black body at 1250 deg. fahr. By using the Stefan-Boltzmann formula, a black body at 1250 deg. radiates about 14,000 B.t.u. per sq. ft. per hr. The tests have also shown that the radiation increases with the thickness of the flame up to about 24 in. Above that no increase in radiation takes place. At this thickness the radiation of the flame would be about 23,300 B.t.u. per sq. ft. per hr. Taking the flame temperature to be about 3300 deg. fahr., the radiation from a black body at this temperature would be about 320,000 B.t.u. per sq. ft. per hr. Therefore, the maximum radiation from a Bunsen flame is only about

$$\frac{23{,}000}{320{,}000} = 0.073$$

or 7.3 per cent of that of a black body at the same temperature.

With a clean luminous flame without noticeable particles of fuel, the radiation is increased about 1.6 to 1.8 times the above. This would make the radiation from the luminous flame, roughly, about 12 to 13 per cent of that from a black body. The radiation from

TABLE 1 FURNACE TEMPERATURES IN DEGREES FAHRENHEIT NEAR WATER-COOLED WALLS

Location	Boiler Rating (Approx.) 300 Per Cent With About 15 Per Cent CO_2 (Distance couple projects into furnace, in.)						Boiler Rating (Approx.) 500 Per Cent With About 15 Per Cent CO_2 (Distance couple projects into furnace, in.)					
No.	6	12	18	24	36	48	6	12	18	24	36	48
1	1520	1760	1800	1850	1900	2020	1570	1640	1780	1770	1920	2020
2	1730	1820	1890	1930	1970	2020	1530	1610	1730	1870	2020	2160
3	1500	1600	1690	1780	1940	2000	1750	1810	1880	1990	2150	2290
4	1640	1700	1760	1820	1900	1920	1850	1940	1960	1990	2050	2300
5	1500	1620	1700	1730	1750	1760	1700	1860	1870	1870	1890	1900
6	1670	1800	1840	1910	1990	2070	1690	1790	1820	1870	2050	2150
7	1540	1640	1700	1750	1800	1830	1640	1840	1900	1870	1940	1990
8	1430	1540	1610	1680	1980	2070	1670	1740	1740	1830	1940	2040
9	1580	1660	1730	1770	1830	1890	1780	1900	1930	1950	1970	2130
10	1380	1410	1460	1550	1630	1700	1590	1570	1650	1650	1770	1850

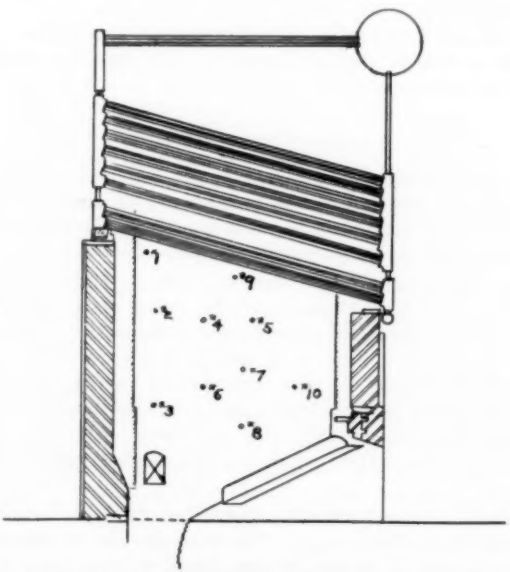

FIG. 1 LOCATION OF PYROMETERS FOR TEMPERATURES OF TABLE 1

the luminous flames in an ordinary boiler furnace is probably considerably greater than that from a clear luminous gas flame at the same temperature because the furnace flames are likely to contain a certain amount of coal or soot particles which would make the flame a more effective radiator. Assuming a furnace of

cubical shape filled with flame and with a grate at the bottom, the surface of the flame would be just five times that of the grate, which would mean that the radiation from the flame is about 60 to 65 per cent that of the fuel bed. This shows that the radiation of the flame is an important factor in computing the heat absorbed in a boiler furnace.

26 Temperature readings taken in a furnace of a large boiler with water-cooled walls have shown the temperature to be lowest at the wall and gradually increasing toward the center of the furnace for about 4 to 5 ft., depending upon the rating of the boiler. While one can hardly expect that the flame for the furnace gases is transparent to its own radiation for more than 3 ft., there is undoubtedly, due to the difference in temperature, an interchange of the gases near the cool wall with the neighboring gases, so that the radiating heat absorbed by the wall affects a considerably greater gas thickness. This explains the fact that the temperature inside of the furnace near the wall is appreciably higher than one would expect, taking into consideration the heat absorbed or the water evaporated by the water tubes.

27 Table 1 shows the temperature in a furnace near water-cooled walls at 300 and 500 per cent of rating, while Fig. 1 indicates where the temperatures were taken.

Heat Absorption by Radiation

28 The heat absorbed by radiation per square foot of surface depends upon the amount of fuel burned and the amount of heating surface that is located in the furnace, both of which influence the temperature of the flame or gases in the furnace.

29 Fig. 2 shows how the temperature of the flame varies with the amount of coal burned, also the influence of water-cooled walls upon this temperature. It is based on a furnace 18 ft. long, 18 ft. high, and 24 ft. wide, fired with coal having a calorific value of about 14,000 B.t.u. per lb., and burned with 14 per cent of CO_2. The first two lower rows of the boiler present about 700 sq. ft. of surface absorbing heat by radiation, and there is also heat-absorbing surface of about 1000 sq. ft. in the water-cooled walls and bottom.

30 Curve A represents the calculated furnace temperature which would result from combustion with the above coal without any losses. Curve B represents the actual average temperatures in the furnace under the above conditions with different rates of fuel consumption. Curve C shows what the temperature would be if no water-cooled radiation surface were present to absorb it. The shaded portion between curves B and C represents the temperature drop in the furnace due to heat absorption by radiation. Curve B approaches curve C as the rate of combustion of fuel increases, which shows that the heat absorbed by radiation does not increase in the same proportion as the fuel burned.

31 Haslam, Lovell, and Hunneman also made a study[1] to determine the effect of flame size and depth upon the radiation per unit area, that is, to ascertain if a flame is transparent to its own radiation, and found that radiation from gases does not follow the fourth-power law of Stefan and Boltzmann. Their experiments confirmed the theory that even a clear non-luminous flame above a certain depth becomes opaque to its own radiation, the critical depth being about 3 ft. The author was unable to find a statement by investigators as to whether or not a flame of a greater thickness is transparent to radiation from other bodies. It is, however, reasonable to assume that this is not the case. In the treatises on radiation by Gerbel and by Roszak and

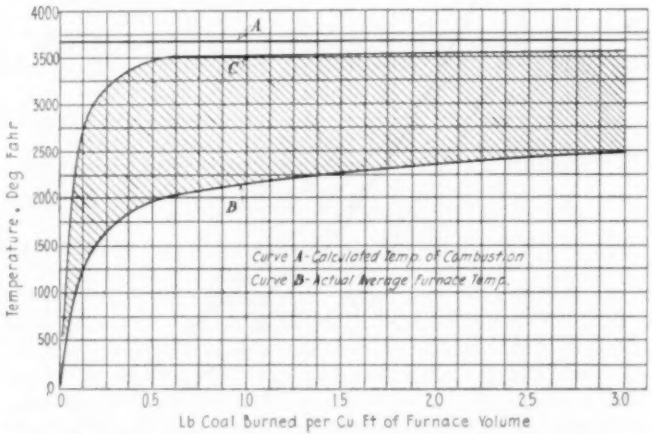

FIG. 2 COMPARISON OF ACTUAL AND THEORETICAL FLAME TEMPERATURES AT DIFFERENT COAL RATES

Veron previously mentioned, the calculations were based on radiation from the fuel bed, but not taking into consideration the effect of the flame on this radiation. The author is also inclined to question the correctness of the assumption of Wohlenberg and Morrow that the radiation in a pulverized-fuel furnace is influenced considerably by the fineness of the coal particles.

32 Probably due to these difficulties of the analytic methods, many engineers have endeavored to find a practical relation between the amount of fuel burned, the surface exposed to direct radiation in the furnace, and the heat transfer per hour per square foot of this surface. In discussing the paper by Wohlenberg, Geo. A. Orrok mentions an empirical formula which was established as far back as the end of the last century. Since the introduction

[1] *Industrial and Engineering Chemistry*, March, 1924.

of water-cooled walls this problem has awakened interest anew, and is the subject of study by those interested in furnace design.

33 The most important factor in connection with heat radiation is the relation between the total heat generated in the furnace and that absorbed by radiation; in other words, the fraction of generated or liberated energy which is absorbed in the furnace. Wohlenberg calls this the furnace absorption efficiency, and designates it by the symbol μ_4. The author endeavored to figure out this relation for a few cases, either directly where the amount of

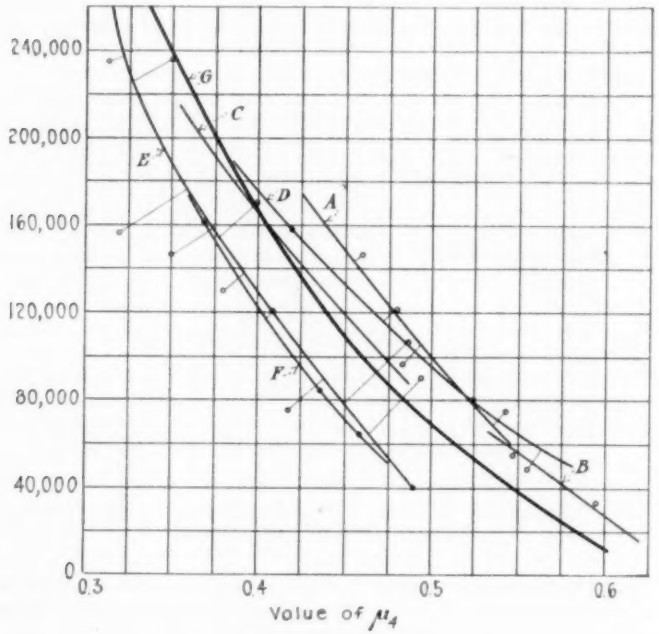

FIG. 3 VALUES OF μ_4

steam generated by the radiant-heat-absorbing surfaces was known, or indirectly by comparison with similar cases where no radiation occurred. There seems to be a certain relation between the value of μ_4 and the ratio between the total heat generated in the furnace and the amount of surface exposed to direct radiation. In Fig. 3 curves are plotted giving μ_4 for different cases. The ordinates show the ratio between the total heat liberated and the surface exposed to radiation, while the abscissas show the values of coefficient μ_4. Curve A is based on the tests with water-cooled walls in one of the original installations at the Cahokia station where there are 1800-hp. boilers fired with pulverized fuel and

having vertical water tubes in the rear wall. Curve B is based on tests at Sherman Creek with a 600-hp. Springfield boiler fired with pulverized fuel and provided with two fin-type water-cooled walls at the sides and a radiant-type superheater at the rear wall. Curve C shows tests made by Weber in Vienna on a water-tube boiler fired with coal on hand-fired grates.

34 While there is no method of figuring out the amount of heat absorbed by the water-cooled walls in the Hell Gate Station, the author endeavored to determine this by working back from the coal burned and the temperature of the exit gases. In this way was determined the heat which would be absorbed in this boiler by convection alone. The remainder was then accepted as being absorbed by radiation, from which μ_4 was determined. The results are plotted as curve D.

35 Curve E is based on tests by W. F. M. Goss with a Jacobs-Shupert locomotive boiler at Coatesville, Pa., between 1911 and 1912. The values of μ_4 presented by curve F are taken from Wohlenberg's paper.

36 It is interesting to note that for a certain ratio between the total steam liberated and the amount of surface exposed to radiation, the variation of μ_4 for the six cases mentioned is less than 20 per cent, which would indicate that the average value of μ_4 as presented by curve G is sufficiently accurate for practical purposes to be used as a standard on which to base calculations. By means of this standard curve, one can determine what part of the total heat generated will be absorbed by the radiating surface in the furnace for a certain rate of operation of the boiler. This can be found by determining the ratio between the total heat to be absorbed at that rate of operation and the total amount of radiating surface. The total heat should be multiplied by the factor μ_4 corresponding to the determined ratio, which gives the total heat absorbed by radiation. This curve can also be used to find the heat transmission by radiation per square foot of heating surface for any given condition of the furnace. Determine the total amount absorbed by radiation as before, and divide it by the number of square feet of surface exposed to radiation.

37 Very often the question arises as to how much steam will be evaporated by the water-cooled surface located in the furnace, that is, the water walls and bottom screen. That this problem can be easily solved by the use of the standard curve G of Fig. 3, is evident.

38 By the total heat-absorbing surface mentioned above is meant the projected surface exposed to the furnace, including the surface of the lower rows of tubes of the boiler section, the side walls of the furnace, and its bottom. As the lower rows of boiler tubes, the lower sides of which are exposed to radiation, are always covered by water, they present an excellent heat-absorbing surface. It has been suggested, therefore, when calculating the total radiant

surface, that 40 per cent of the lower tube surface be added or the tube surface be multiplied by 1.4. More correct results are also obtained if the projected surface of the bottom screen tubes is multiplied by 0.75 to take care of the fact that this surface is mostly covered by ashes which reduce its heat-absorbing capacity.

39 Table 2 shows the heat transfer per square foot of projected area of the side wall of a standard water-tube boiler of about 1600 hp. for different percentages of radiant surface and amounts of coal burned.

40 The heating surface of a boiler tube is usually considered its full circumference, irrespective of the way the gases are passing over it. The heating surface of a water-cooled wall absorbing heat by radiation is given in different ways. The heat absorbed per square foot of surface depends upon the way the heating surface is considered. Fig. 4 is based on 800 sq. ft. of water-cooled furnace wall and a boiler with 480 sq. ft. of projected area exposed to

TABLE 2 B.T.U. TRANSFER PER SQ. FT. OF PROJECTED AREA PER HOUR TO FURNACE-WALL SURFACE OF STANDARD WATER-TUBE BOILERS

Millions B.t.u. liberated per hour in furnace	Furnace-wall surface in sq. ft.					
	250	500	1,000	1,500	2,000	3,000
50	29,300	21,800	16,400	13,600	11,400	8,400
100	49,500	41,000	31,000	24,600	20,500	15,200
150	66,000	55,500	42,200	34,200	29,500	22,300
200	77,200	67,300	52,600	43,400	36,700	28,600
250	90,700	79,000	61,800	51,000	44,000	34,000
300	98,600	86,200	69,400	58,500	51,000	39,200

This table is based on boiler having 480 sq. ft. of projected area exposed to radiation in the furnace.

radiation. The heat absorbed by radiation depends upon the amount of coal burned. The heat transfer per square foot per hour, however, depends upon how the surface is considered. If we assume the whole circumference of the tube to be heating surface, the maximum heat transfer with 24,000 lb. of coal burned per hour is 24,500 B.t.u. per sq. ft. If only the projected area is considered as heating surface, the maximum heat transfer for the same conditions is 77,000 B.t.u. per sq. ft. If half of the circumference is taken, or only the area of the tube which "sees" the furnace, the heat transfer is as shown on curve C, while if the projected area is assumed as being the heating surface, curve D indicates the heat transfer.

41 In order to avoid misunderstanding in speaking of water-screen surfaces, it may be advisable to establish a standard for denoting which surface is to be used in specifying the number of square feet of effective heating surface of a water screen for a boiler furnace. The author would suggest that the projected area be used, which for tubes is equal to the tube diameter multiplied by the length exposed.

Most Advantageous Amount of Radiant-Heat-Absorbing Surface in a Furnace

42 In order to obtain the highest furnace temperature and the correspondingly highest efficiency, it is desirable to have the most

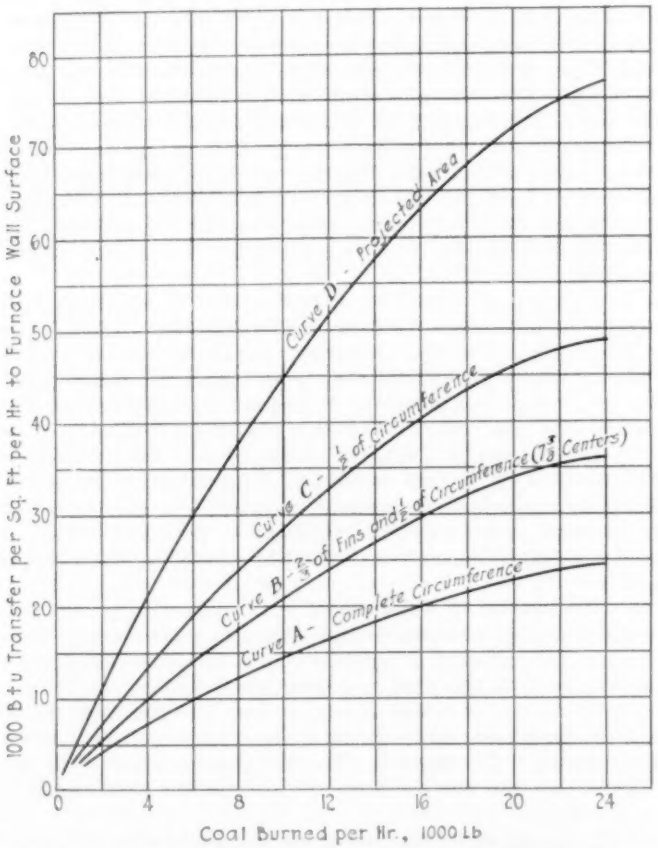

Fig. 4 Comparison of Different Methods of Computing Evaporation of Side-Wall Surface

(Curves based on 800 sq. ft. of furnace-wall surface being water-cooled and on boiler having 480 sq. ft. of projected area exposed to radiation in the furnace.)

rapid ignition and combustion possible. The coal entering the furnace must first be heated, and any moisture in the coal must vaporize before combustion can begin. The burning gases must furnish the heat to raise the temperature of the coal and to evaporate the moisture. Some of this heat is absorbed by radiation

from the surrounding walls if they are not water-cooled, but even in this case the greater part comes from the flames of the burning gases. When the coal is heated up to about 400 deg. fahr. the gases and volatile vapors are given off from the coal before combustion starts.

43 Coal, being a black body, readily absorbs radiant heat, thus increasing in temperature. The gases of distillation also absorb more heat by radiation. After the gases and the admitted air have reached the ignition temperature, the combustion of the gases commences. The burning volatile matter produces water vapor and carbon dioxide, both of which will absorb and also radiate a considerable amount of radiant energy. Experiments have shown that luminous flames with particles of carbon floating in them radiate more heat than non-luminous flames. With pulverized fuel more radiant heat may be absorbed from the flames than with underfeed stokers, due to a larger volume of flame of greater luminosity.

44 On the other hand, in a pulverized-fuel furnace such solid particles as coke, burning carbon, and incandescent ash are in the path of radiation from the interior portions of the flame, and either absorb or reflect this radiant heat. It is logical to assume, therefore, as already mentioned in a previous paragraph, that the furnace walls affect only the radiation of a relatively thin envelope of the furnace flame, and that the interior of the furnace is affected but little, if at all, by wall conditions. Many examples prove this. This is of considerable importance, as it tends to show that wall temperatures cannot affect the conditions in the whole furnace, a question which is in everybody's mind in considering water-cooled walls.

45 The amount of water-cooled surface that can be installed without lowering the furnace temperature to a point where combustion will be affected is dependent upon the kind of coal used, particularly upon the kind and quantity of volatile matter and of ash in the coal, and also upon the amount of moisture. All of these are important factors in the ignition temperature and affect the inflammability of the coal. The fusing temperature of the ash is also to be taken into consideration with respect to the slagging of the tubes.

46 In regions where only low-grade coal is available, difficulties have been experienced with the furnace walls, even when air-cooled. The refractory is eroded rapidly by the fused ash. Air cooling of the side walls becomes effective only when the thickness of the refractory is somewhere between one and two inches, which is difficult to maintain. It is therefore very desirable when low-grade coal is used to keep the walls cool by water tubes. Enough refractory must be in the furnace, however, to reflect sufficient heat so that the temperature of the gases throughout the entire furnace is not lower than that required for combustion. The

opinions of engineers vary widely regarding the amount of refractory or brick surface required in a furnace burning low-grade coal. Many believe that brick walls are needed as reflecting surfaces to accelerate the ignition and combustion of the coal, and that water-cooled walls would absorb too much radiant heat and cause incomplete combustion. As shown above, only the outer envelope of the furnace flames is affected by the cooling action of the tubes. If mixing of the air and gases could be maintained so that there would be an interchange between the gas near the walls and that in the center of the furnace, the ill effect of the water tubes could be minimized considerably. Preheated air would also reduce this effect.

47 It is usually thought that water-cooled walls should be used with pulverized fuel and that they are less necessary with stokers. We need only consider the fact that in stoker furnaces pressures higher than the atmosphere occur, which have a very bad effect on the brickwork, particularly if the temperature is very high, to understand that from this point of view water-cooled walls are more beneficial for stoker-fired than for powdered-coal furnaces.

48 Investigations of the burning of coal in furnaces have shown that anything that will burn at all will ignite at an ordinary furnace temperature of about 2000 deg. fahr.; that the question of combustion is not how to combine the oxygen and carbon when brought together, but how to bring them together. At ordinary furnace temperatures, the speed of the reaction between oxygen and carbon is so great that these elements will combine practically instantaneously. The main problem is to bring the particles together or to mix the oxygen and carbon, which requires time and space. Temperature has little effect on the mixing. It seems, therefore, that the furnace temperature can be reduced by heat-absorbing surface without affecting the combustion, if means are provided for the proper mixing of air and fuel.

49 Water-cooled furnaces naturally permit higher boiler capacities for the same boiler and the same furnace volume, as the rate of heat absorbed by the water-cooled tubes in the furnace is very high. Where water-cooled walls are used and the average temperature of the furnace is lower, no trouble is experienced with the erosive effect of the gases, and probably little trouble will be experienced with slag. The average higher capacity per square foot of heating surface would enable one to design the boiler proper with less heating surface, which would result also in reducing brickwork as well as height of the boiler and boiler room.

50 The greater the amount of moisture in coal, the more heat must be absorbed before the coal is elevated to the ignition temperature. This applies also to the amount of ash in the fuel. On the other hand, the greater the volatile hydrocarbon content of the coal, the easier it will ignite. These naturally have an effect on the amount of cooling surface that can be arranged in a furnace.

51 The greater the amount of volatile matter, the more heating surface can be installed in the furnace; but the greater the ash or moisture content, the higher is the furnace temperature required, and the more dependent we are upon the reflected heat from the brick, with the result that less water-cooled surface should be provided. Where coal with ash of a low fusing point is used, water-cooled walls reduce the clinkering and slagging.

52 There is one factor in connection with water-cooled walls which does not seem to have been given any attention. It is a fact that the water-cooled furnace lacks the heat-storage capacity which is usually present in a brick furnace. A furnace with a large heat-storage capacity does not lend itself well for control of furnace operation. On the other hand, however, it enables the rate of operation to be rapidly increased when so desired. This may be of considerable importance for power stations with high peak loads.

53 To sum up: Where coal which has a moisture content not greater than 4 per cent, a volatile content of approximately 20 per cent, and an ash content of not more than 7 or 8 per cent is burned in pulverized form, there is no reason why the furnace cannot be completely covered with water-cooled surfaces. For the same coal burned on stokers it would be advisable to have the lower part of the furnace lined with brick in order to accelerate combustion on the stoker. With coal containing moisture up to 10 per cent and ash from 7 to 10 per cent, about 20 to 25 per cent of the surface will have to be of refractory unless means are provided for effectively mixing the gases in the furnace. It appears, however, that with coal having 25 per cent of moisture or a very high ash content it is not advisable to have more than two sides of the furnace water-cooled; when burned in pulverized form the bottom may also be water-cooled. In some cases it will be necessary also to fill in the spaces between the tubes of the side walls with brick. With preheated air the amount of cooled surface can be increased in every case.

54 The above is merely the author's opinion based on observations of the very limited number of installations having this type of furnace. Many engineers will have other views regarding the cooling of furnace walls.

55 There is a noticeable tendency toward large furnaces with a maximum possible amount of radiant-heat-absorbing surface, particularly where coal with low moisture and ash contents is used. Most of the steam is generated by this surface in the furnace, and the gases are there cooled to a temperature at which no more heat can be absorbed by radiation than by convection. They pass then to a comparatively small boiler section where the remainder of the steam is generated. They are usually provided with an air heater to raise the air for combustion to high temperature. It is understood that such a boiler is now being built in Great Britain.

56 Fig. 5 shows a similar boiler designed by Mr. Burroughs of the Canada Sugar Refining Company, Ltd., in Montreal. This furnace has the distinction of being circular, so that each tube absorbs an equal amount of heat. In the opinion of the author a better design would be to have the tubes inclined inward at the top. The advantage of this is that the sides of the tubes toward the furnace will always be covered with water as the forming steam bubbles, having a tendency to rise, will move along the rear sides of

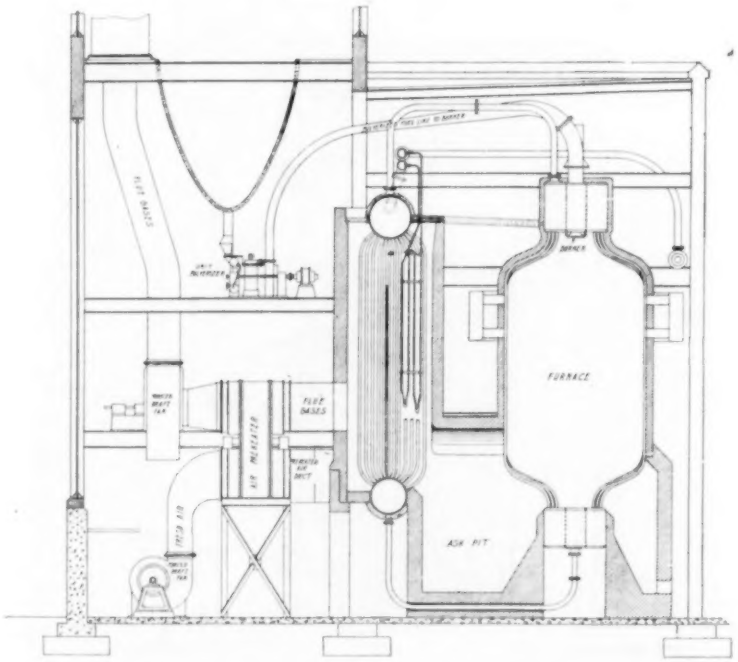

FIG. 5 LAYOUT OF BOILER WITH SEPARATE CIRCULAR WATER-COOLED FURNACE DESIGNED BY MR. BURROUGHS FOR THE CANADA SUGAR REFINING COMPANY, LTD.

the tubes. It also makes it possible better to secure the insulating material to the tubes of the water-cooled walls. The inclined position of the tubes would tend to retard somewhat the too rapid circulation and maintain a steadier water level in the drum.

57 There are different arrangements of water-cooled walls. Fig. 6 shows a stoker-fired boiler with side walls as installed in the Hell Gate Station of the United Electric Light & Power Company, New York. The wall consists of 4-in. tubes arranged vertically on each side of the furnace and spaced on about 7-in. centers. Each

tube has two longitudinal steel fins welded on it diametrically opposite each other, and when placed in the boiler furnace the fins of the adjacent tubes butt up against each other or overlap, thus

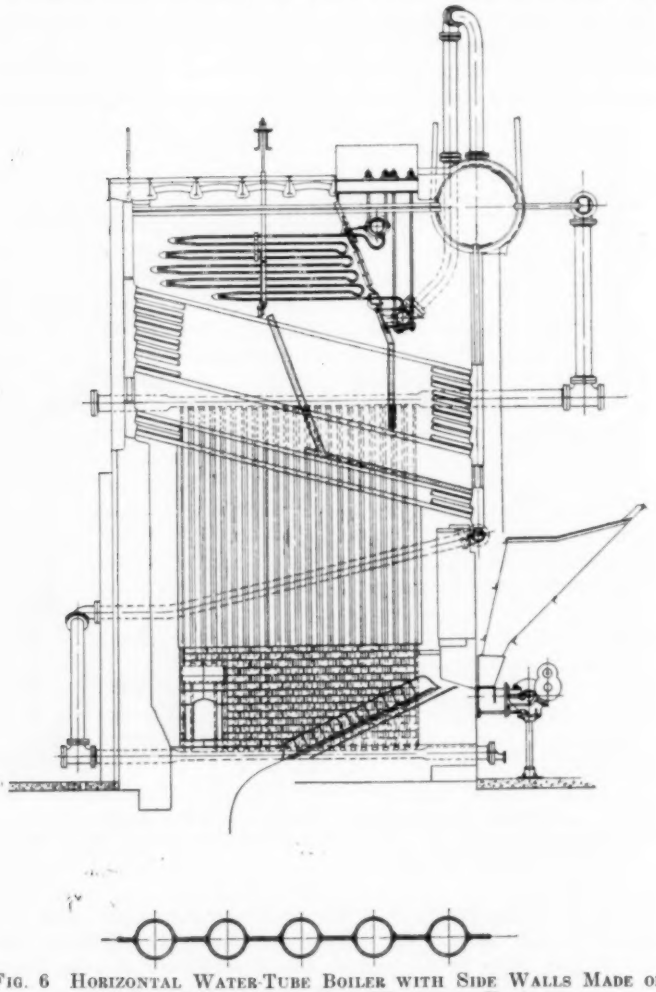

FIG. 6 HORIZONTAL WATER-TUBE BOILER WITH SIDE WALLS MADE OF FOUR-INCH TUBES WITH WELDED-ON FINS

presenting a continuous water-cooled surface to the radiant heat of the furnace. The lower portion of the tubes is covered by fire-clay tile for a short distance above the stoker, and these extend along a horizontal line the entire length of the furnace.

58 Fig. 7 illustrates an installation of a semi-vertical boiler with water-cooled walls at both sides. The water-cooled surface consists of bifurcated tubes which cover the wall completely.

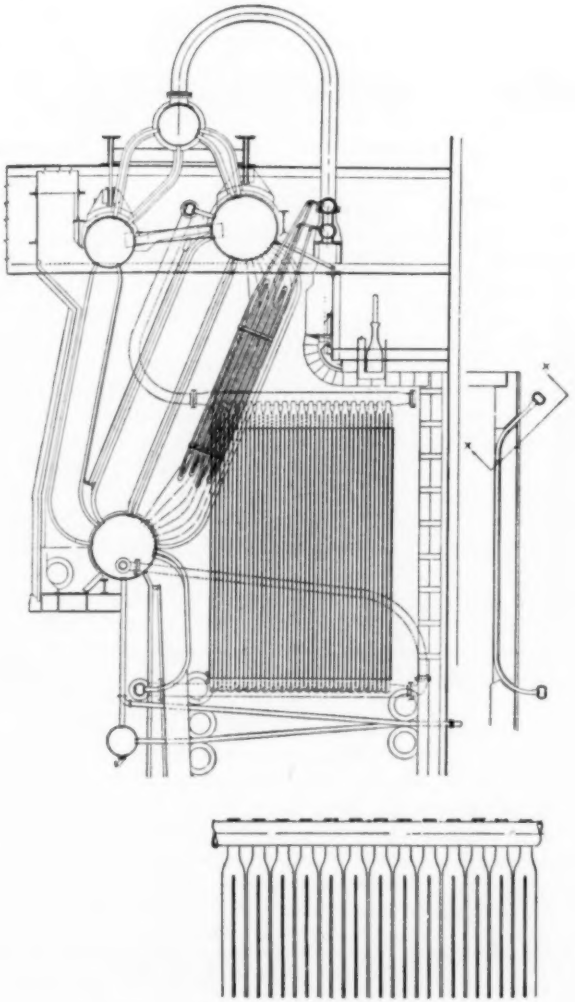

FIG. 7 SEMI-VERTICAL BOILER WITH WATER-COOLED WALLS CONSISTING OF BIFURCATED TUBES

59 Fig. 8 indicates another design of water-cooled furnace walls. These walls, in accordance with the statement of the manufacturer, consist of steel tubes on which are shrunk heavy rectangular cast-

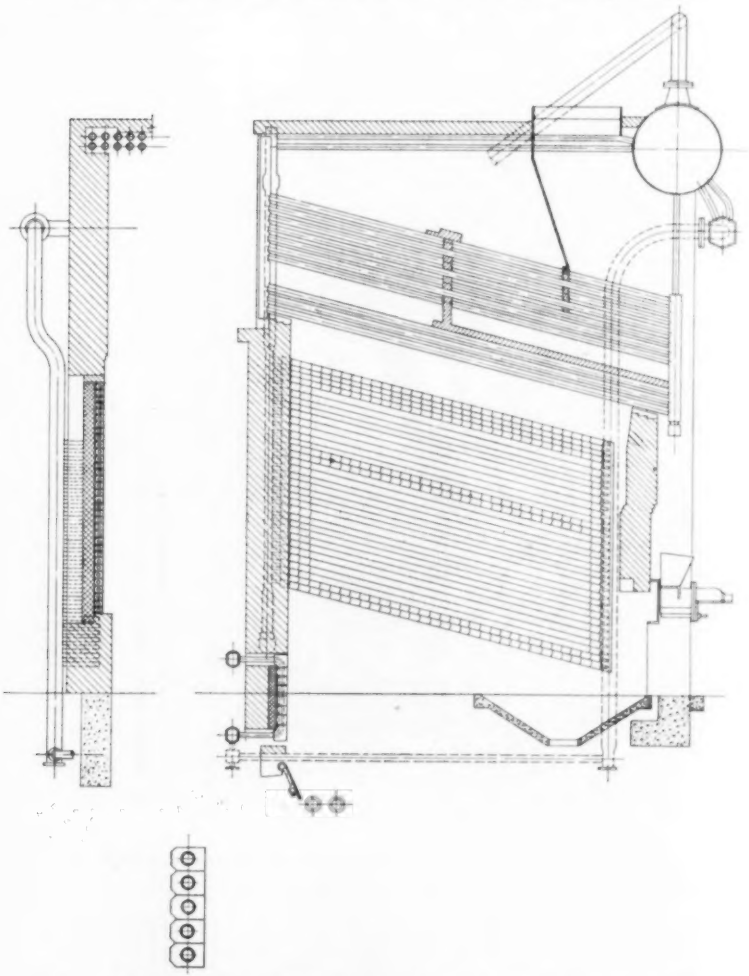

FIG. 8 WATER-COOLED WALLS CONSISTING OF STEEL TUBES ON WHICH ARE SHRUNK CAST-IRON BLOCKS

iron blocks about 6 in. long and 4 in. wide. These elements are installed on close centers so that the open space between adjacent elements is approximately $\frac{1}{8}$ in. The faces toward the fire form a continuous flat wall.

60 Another design has been suggested consisting of steel tubes imbedded in the wall, with cast-iron blocks fastened to them facing the furnace. These blocks can be partially covered with refractory material in order to reduce the surface exposed directly to radiation.

61 Some of the above designs of water-cooled walls have heat-absorbing surfaces exposed to the furnace but not cooled directly by water. For these cases a certain factor must be introduced, depending upon their design, in order to determine the heat absorbed by radiation when a value μ_4 of Fig. 3 is used, as these values apply to surfaces directly cooled by water.

Effect of Gas Radiation on Heat Transmission

62 Tests with steam boilers have shown that the coefficient of heat transmission per square foot per degree of temperature difference per hour varies with the temperature difference, and is considerably higher at higher than at lower differences. It was rather difficult to understand this phenomenon, as one would expect from the nature of heat transmission by convection that at higher temperature differences the coefficient would decrease. However, it has been conclusively proved recently that the increase of heat transmission with the higher temperature difference is due to the radiation of gases.

63 Dr. Alfred Schack has made some very interesting investigations at the Heat Research Bureau of the Iron and Steel Institute in Düsseldorf, Germany, and has published his results in its Bulletin 55. The results show which gases are radiating heat and to what extent. It may be of interest to learn how these data were obtained. In accordance with Kirchhoff's rule, the radiation of a body at a given temperature is proportional to its ability to absorb heat. Heat as well as light is ether motion, the difference between the two being only in the wave lengths.

64 By means of a spectrum the light rays and also the heat rays absorbed by certain bodies can be determined. In this way, therefore, can be determined the heat absorbed by gases, which, in accordance with Kirchhoff's rule, will show also the amount of heat the gas is radiating. The absorption spectra of several gases which are the chief constituents of boiler flue gases, namely, nitrogen, carbon dioxide, oxygen, and water vapor, show that nitrogen and oxygen have no appreciable absorption, while carbon dioxide and water vapor have absorption spectra of comparatively large extent. Combustion gases which contain carbon dioxide and water vapor can radiate considerable heat even if they are non-luminous and scarcely visible.

65 The spectrum also shows that there is a definite relation between the energy radiated, wave length, and temperature; also that the thickness and shape of the body of gases influence the

radiation. In his bulletin, Dr. Schack gives equations and tables which enable one to compute the radiation of gases. As the study of the radiation of gases is of considerable interest in connection with steam boilers, open-hearth furnaces, etc., the method of computing the radiation of gases is shown in the Appendix.

66 Of particular interest is the effect of radiation of gases in connection with the heat transmission in a steam boiler, which will be shown by the following example.

67 Two points of the path of the gases in the boiler were selected where the temperatures were 1400 and 800 deg. fahr., respectively. From experiments, the heat transferred with 6000 lb. of gases per sq. ft. of area will be about 7.7 B.t.u. per deg. temperature difference per hr. at the 1400-deg. point. With 250 lb. pressure the temperature of the water is about 406 deg. The metal temperature of the tubes follows very closely the temperature of the water, so that the temperature difference between the tubes and the gases will be 994 deg., or a total heat transmission of 7660 B.t.u. per sq. ft. per hr. At the 800-deg. point the heat-transfer rate is 6.65 with a temperature difference of 394 deg., giving a heat transmission of 2620 B.t.u. per sq. ft. per hr. Assuming the flue gases to contain 15 per cent CO_2 and 6 per cent H_2O, and the thickness of the gas layer to be about 6 inches, and using Equations [**1**] and [**3**] of the Appendix, the following table of values at 1400 and 800 deg. may be computed:

	1400 deg.	800 deg.
Total B.t.u. transmission per sq. ft. per hr	7660	2620
B.t.u. radiated by CO_2 (Equation [**1**]) per sq. ft. per hr.	1340	280
B.t.u. radiated by H_2O (Equation [**3**]) per sq. ft. per hr.	477	107
B.t.u. radiated by CO_2 plus H_2O per sq. ft. per hr	1817	387
B.t.u. by convection only, line 1 minus line 4	5843	2233
B.t.u. transmitted by convection only per sq. ft. per hr. per deg. temperature difference	5.88	5.66

68 The figures indicate that it may be safely assumed that heat radiation is a considerable factor in heat absorption by a boiler. It becomes necessary to consider the heat transfer from gases as consisting of two components, one following the law of convection and the other the law of radiation. As the heat transmitted by convection varies with the velocity of the gases while the radiant heat is independent of this factor, it is important to have high velocity of the gases only where the heat absorbed by convection is dominating. This is of considerable interest in determining the sectional area of the gas flow in a boiler. An increase in velocity of the gases will increase the heat transmission only in the last passes of the boiler where the radiation is very low and most of the heat is transmitted by convection, so that it is advisable to have the gases pass over the heating surface with the highest possible velocity and in streams of small area. In the passes nearer

to the furnace where the gas temperatures are high, it is rather desirable to have the gases flow with low velocity and in streams of large area in order to increase the radiant effect and, due to being in the pass a longer time, make it possible to give off a considerably greater amount of heat.

The Influence of Radiation upon the Measurement of Gas Temperature

69 It is a known fact that a thermometer unless properly screened cannot record the true temperature when placed in a furnace in which the walls or some portions of them are at a lower temperature than that of the furnace proper, as, for instance, in a locomotive firebox. In such a case a thermometer bulb radiates and absorbs heat much more readily than the flame and gases in which it is immersed. The cooler walls absorb more radiant heat than they emit, and therefore, if the thermometer can " see " the cold walls it radiates more heat than it receives from these walls, with the result that it tends to fall in temperature.

70 One example of this defective indication of gas temperature is given in Bulletin 23 of the Bureau of Mines with respect to a test upon a Stirling boiler having air-heated tubes near the flue-gas outlet. The difference between the actual and the recorded temperatures may be particularly large in a water-cooled furnace where the temperature differences between the gases or the pyrometers and the water-cooled walls is very great, so that there is an appreciable radiation from the pyrometer. Extensive experiments on different boilers have been made by H. Kreisinger and J. F. Barclay on the influence of radiation in measuring the temperature of boiler flue gases. Two series of hot junctions were compared with each other at different points in each boiler. One junction was made of wires 0.08 in. in diameter and the other made as a tube 0.5 in. in diameter. By extrapolation the experimenters also estimated the true gas temperature. The variations obtained with the thick junction indicate that a large error may be incurred by using the pyrometer enclosed in a tube of 1½ in. diameter if not shielded from the radiation from the cold tube surface. This error can be reduced by arranging a small screen of asbestos or refractory material around the bulb or junction, without unduly preventing the gases from coming into true contact with it.

APPENDIX

71 In order to compute the radiation of gases, Dr. Schack gives in Bulletin 55 of the Heat Research Bureau of the Iron and Steel Institute Düsseldorf, Germany, the following equation, [1];

$$Q = \frac{C}{4.9}\left[a(K_1 - K'_1)(\mathrm{V}) + (K_2 - K'_2) + a(K_3 - K'_3)(\mathrm{II})\right] \quad . \quad . \quad [1]$$

for the radiation of CO_2 in calories per square meter per hour, in which C is the radiation coefficient for the surface absorbing heat and is 1480 to 1550 for brick or iron or 4.0 to 4.5 in the metric system. a is a coefficient depending upon the shape of the radiating gas body. For

TABLE 3 VALUE OF COEFFICIENTS FOR RADIATION OF CO_2. RADIATION IN CAL. PER SQ. M. PER HOUR OF AN INFINITELY THICK LAYER OF CARBON DIOXIDE

		Radiation of		
Temp. deg. cent.	Band 1 K_1	Band 2 K_2	Band 3 K_3	Black Body
200	$0.004 \cdot 10^3$	$0.007 \cdot 10^3$	$0.23 \cdot 10^3$	$2.4 \cdot 10^3$
300	0.045 "	0.24 "	0.36 "	5.2 "
400	0.15 "	0.52 "	0.50 "	10.0 "
500	0.42 "	0.95 "	0.65 "	17.6 "
600	0.96 "	1.6 "	0.79 "	28.4 "
700	1.9 "	2.4 "	0.90 "	42 "
800	3.2 "	3.4 "	1.1 "	64 "
900	4.8 "	4.3 "	1.3 "	90 "
1000	6.8 "	5.4 "	1.4 "	125 "
1100	9.3 "	6.7 "	1.6 "	152 "
1200	12.3 "	8.3 "	1.9 "	230 "
1300	15.6 "	9.7 "	2.2 "	300 "
1400	19.2 "	11.2 "	2.4 "	385 "
1500	23.4 "	12.9 "	2.7 "	490 "
1600	27.9 "	14.6 "	3.0 "	600 "
1700	33 "	16.6 "	3.4 "	730 "
1800	38 "	18.7 "	3.8 "	880 "
1900	44 "	21.1 "	4.1 "	1050 "
2000	51 "	23.6 "	4.5 "	1290 "

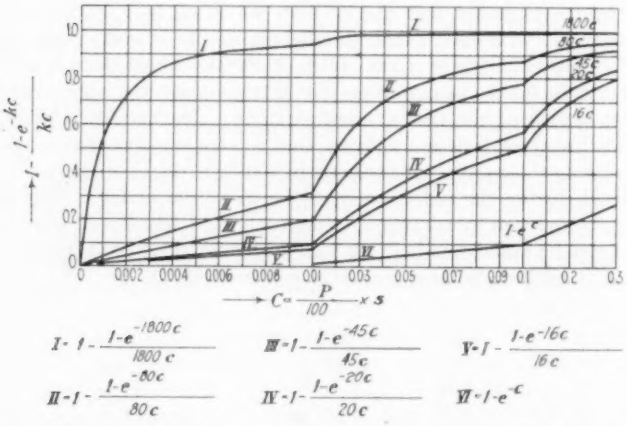

FIG. 9 CURVES SHOWING VALUES OF FUNCTIONS USED IN EQUATIONS FOR RADIATION, [1], [2] AND [3]

compact shapes, as a cube or sphere, $a = 0.7$ to 0.8. For long flat shapes, as ducts, $a = 1.1$ to 1.0. The figures given first are for small values of $\frac{P}{100} \times s$. V and II are functions of a as well as of $\frac{P}{100} \times s$, the values of which can be found in Fig. 9. The symbols K_1, K'_1, K_2, etc., are coefficients denoting the radiation of the various absorption bands, if produced by infinitely thick layers of CO_2. Their values will be found

TABLE 4 VALUES OF COEFFICIENT FOR RADIATION OF WATER. RADIATION IN CAL. PER SQ. M. PER HOUR OF AN INFINITELY THICK LAYER OF WATER VAPOR

Temp. deg. cent.	Band 1 W_1	Radiation of Band 2 W_2	Band 3 W_3	Black Body
200	$0.006 \cdot 10^3$	$0.38 \cdot 10^3$	$0.63 \cdot 10^3$	$2.4 \cdot 10^3$
300	0.075 "	0.88 "	0.81 "	5.2 "
400	0.25 "	1.6 "	1.1 "	10.0 "
500	0.75 "	2.4 "	1.4 "	17.6 "
600	1.5 "	3.6 "	1.6 "	28.4 "
700	2.8 "	4.9 "	1.9 "	42 "
800	4.7 "	6.3 "	2.2 "	64 "
900	5.8 "	7.9 "	2.7 "	90 "
1000	10 "	9.3 "	3.1 "	125 "
1100	14 "	11 "	3.4 "	152 "
1200	18 "	13 "	3.7 "	230 "
1300	23 "	14 "	4.1 "	300 "
1400	29 "	16 "	4.5 "	385 "
1500	35 "	18 "	4.9 "	490 "
1600	41 "	20 "	5.2 "	600 "
1700	48 "	22 "	5.6 "	730 "
1800	56 "	24 "	6.0 "	880 "
1900	63 "	26 "	6.4 "	1050 "
2000	72 "	28 "	....	1290 "

in Table 3. K refers to temperature of the gases, while K' refers to the temperature of the surface receiving radiation.

72 If $\frac{P}{100} \times s$ is less than 0.01, which condition occurs for very shallow ducts, or for less than 2 per cent by volume content of gases, Equation [1] is not accurate enough and is replaced by

$$Q = \frac{C}{4.9} [a(K_1 - K'_1)(\mathrm{V}) + 0.64a(K_2 - K'_2)(\mathrm{I}) + 0.45a(K_3 - K'_3)(\mathrm{II})] \quad . \; . \; . \; [2]$$

the values of I being found in Fig. 9.

73 For water vapor the same procedure holds good as for CO_2 but different numerical values are used. Water vapor also has three absorption bands. The radiation of a hot gas body s meters thick and containing P per cent water vapor is given by the following equation,

$$Q = \frac{C}{4.9} [a(W_1 - W'_1)(\mathrm{IV}) + a(W_2 - W'_2)(\mathrm{III}) + a(W_3 - W'_3)(\mathrm{VI})] \quad . \; . \; . \; [3]$$

The Roman numerals are functions of $\frac{P}{100} \times s$ and are taken from Fig. 9 as for Equation [1] and [2]. Values for W_1, W_2, etc., will be found in Table 4.

74 There are given in the bulletin several examples of the use of the above formulas.

DISCUSSION

GEO. A. ORROK.[1] The absorption of radiant heat in boiler furnaces has long occupied the attention of engineers and physicists, the latter making much the larger number of experiments and producing many papers and reports. In consequence a rather complicated theory has been evolved which depends on the accurate measurement of high temperatures in the furnace, which remains up to date a very difficult proceeding. Such work as that of Nicolson, Callendar, and others, the paper by Wohlenberg,[2] and such books as those of Roszak, Münzinger, and Fry serve but to emphasize the difficulties of any rational method of calculating radiant-heat absorption in such a complex situation as a boiler furnace.

But the problem is still with us and a solution must be reached, empirical if not rational, since the increasing demands on boiler surface have made water-cooled walls, water screens, radiant-heat superheaters, and other devices a practical necessity. The use of powdered coal as a fuel has increased both the need and difficulties. Fortunately among all the experimental work recorded there are a number of experiments in which the radiant surfaces were isoiated, and from these it is possible to secure data on transmission of heat which are sufficiently accurate for an empirical solution over a wide variety of conditions.

LOCOMOTIVE-BOILER RADIANT-HEAT DATA

D. K. Clark, in his book on the steam engine, quotes many experiments where such data were secured. He begins with Nicholas Wood's account of the work of George and Robert Stephenson while developing the boiler for the famous *Rocket* in 1828, followed by other experiments with qualitative results. But he finally quotes the work of Couche and Havrez in 1874 on a Chemin de Fer du Nord locomotive boiler cut into five parts and fed by five feed pumps. Fifteen tests were run, five with coke and five with briquets with all the tubes in service, and five with briquets with half of the smoke tubes plugged. The tests bear internal evidence of careful work and are believed to be substantially correct. Values of C_R, the coal burned per square foot of radiant surface, vary from 5.68 to 13.45, with transmissions of from 23,150 to 51,700 B.t.u. per sq. ft. per hr. W. F. M. Goss, Past-President of this Society, in 1912 reported 21 tests of locomotive boilers where the firebox surface was segregated from the tubes.

[1] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.

[2] Delivered at the Spring Meeting, A.S.M.E., Milwaukee, Wis., May, 1925. See p. 127, ante.

The tests were very carefully made, with all the refinements of modern testing, using two qualities of bituminous coal as well as oil. Here the values of C_R varied from 3.09 to 21.02, with transmissions of from 25,790 to 57,250 B.t.u. per sq. ft. per hr.

Water-Tube-Boiler Radiant-Heat Data

Rowan, in his Modern Steam Boiler, quotes the work of Bertin on a Niclausse boiler in which each bank of tubes was isolated. He gives six averages with briquetted fuel where the values of C_R varied between 2.9 and 17.4 with transmissions of from 11,320 to 47,700 B.t.u. Burstall and Cockburn, in the discussion of Dalby's paper [1] on the Transmission of Heat, quote a test of a Belleville boiler in which each row of tubes was segregated. Here the value of C_R is 6.98 and the transmission is 41,300 B.t.u.

In a report [2] of a test of an Atmos boiler the value of C_R is given as 10.8 and the transmission as 38,300.

Thomas E. Murray, in a recent paper presented to the Association of Edison Illuminating Companies, has given the results of tests at Waterside and Sherman Creek. At Waterside two 4-in. tubes were installed about 6 in. below the lower row of tubes and tested for heat transmission by passing a stream of cold water through them. The observed transfers averaged 56,000 B.t.u. and the coal burned per square foot of radiant surface $= C_R = 11.0$, within a fair degree of accuracy.

A 5-sq. ft. side-wall box was installed on one side of the furnace. At $C_R = 15.2$ a transmission of 66,000 was secured. A 140-sq. ft. side-wall box was installed on another boiler and at $C_R = 7.94$ a transmission of 43,000 was secured. In all of these cases it is assumed that the standard B. & W. spacing allows 496 deg. of surface of the two lower rows of tubes to "see" the radiating surface and the coal per square foot of radiant surface, C_R, is calculated on this supposition. This works out to 1.38 times the surface of the lower row of tubes which are exposed to the radiant action of the furnace.

At Sherman Creek one of the pulverized-coal-fired boilers, provided with Murray fin-tube water walls, radiant-heat superheaters, and water-cooled burner boxes, was used in tests. Each kind of surface was on a separate feed pump except the superheater, and fifteen very accurate tests were obtained. The coefficient for the radiant-superheater surface varied between 0.4 and 0.6, but in the calculations 0.5 was used throughout. In these tests

[1] Institution of Mechanical Engineers, Proc., 1909, Parts 3-4, p. 1045-1046.

[2] *Zeitschrift des Vereines deutscher Ingenieure*, vol. 69, no. 7, Feb. 14, 1925, p. 169-173.

the values of C_R varied between 3.38 and 8.21, with transmissions between 20,900 and 48,500. Two kinds of pulverized bituminous coal were used.

Discussion of the Data

We now have data on three designs of locomotive boilers, four designs of water-tube boilers, and one design of revolving boiler, the Atmos. We have data on high- and low-volatile bituminous coal, briquets, coke, powdered fuel, and oil as fuel, with hand-fired, stoker and other types of firing mechanisms. These are given in Table 5. From such a collection of data it should be

TABLE 5 DATA FOR RADIANT HEAT

	S_r	C_R	N
Robert Stephenson about 1834 D. K. Clark			
Coal	6	7.67	15,500
Chemin de Fer du Nord, Couche and Havrez D. K. Clark and Rowan			
Coke	76.5	5.71	23,150
	76.5	8.55	30,400
	76.5	9.53	33,600
	76.5	10.4	33,700
	76.5	10.1	27,300
Briquets	76.5	6.25	27,200
	76.5	9.72	35,500
	76.5	12.1	44,200
	76.5	13.45	49,600
	76.5	12.8	45,000
	68.0	5.68	30,600
	68.0	8.95	34,800
	68.0	10.35	45,800
	68.0	11.6	50,500
	68.0	12.4	51,700
Goss, Jacobs-Shupert, and Radial Firebox			
Bituminous Coal	230.8	5.93	25,840
	230.8	7.18	29,190
	230.8	11.16	37,600
	230.8	13.39	41,290
	230.8	18.82	50,360
	230.8	19.35	50,600
	206.7	8.51	33,900
	206.7	7.03	29,950
	206.7	11.96	36,300
	206.7	14.17	46,200
	206.7	21.02	57,250
	206.7	20.68	51,660
Oil	230.8	3.09	25,790
	230.8	6.10	43,810
	230.8	5.87	38,790
	230.8	9.05	49,140
	230.8	8.88	47,100
	206.7	3.32	32,450
	206.7	6.32	45,750
	206.7	6.05	51,360
	206.7	8.85	56,040
Niclausse Boiler, Bertin			
Briquets	74.2	2.9	11,320
	74.2	5.8	22,100
	74.2	8.7	30,000
	74.2	11.6	36,900
	74.2	14.5	42,700
	74.2	17.4	47,700

	S_r	C_R	X
Belleville Boiler, Burstall and Cockburn			
Coal	85	6.98	41,300
Atmos Boiler, V. D. I.			
Coal	140	10.8	38,300
Murray, Waterside			
Coal	352.8	11	56,000
	251	15.2	66,000
	479	7.94	43,000
Murray, Sherman Creek No. 10			
Powdered Coal	966	3.38	22,225
	966	3.38	20,900
	966	3.77	24,800
	966	3.92	25,225
	966	3.94	23,600
	966	4.22	24,800
	966	4.7	29,500
	966	5.14	32,050
	966	5.38	30,525
	966	5.7	32,700
	966	6.8	43,700
	966	8.21	44,500
	966	7.73	42,700
	966	7.95	48,500
	966	8.1	44,400

possible to work out a formula which would permit of results being predicted with a reasonable degree of accuracy and at the same time permit of the checking up of boiler tests when the radiant surface was not isolated from the circulatory system of the boiler and which at the same time would allow for the determining of the convection fraction of the heat and the temperatures which might be observed in the tube bank and uptakes. A search through the literature of radiation revealed many theoretical attempts to solve the problem (without actual data), and one empirical attempt to check results which seemed to offer promise. This was the work of J. G. Hudson published in *The Engineer* (London), December 26, 1890, and reprinted in Bryan Donkin's Boiler Tests and Rowan's Modern Steam Boilers. Hudson's formulation was rather involved, but could be simplified when certain additional data were known. His theory was that the transmission by radiation was proportional to the coal burned per square foot of radiant surface, and that the furnace temperature was inversely proportional to the pounds of air per pound of fuel.

Let S_t = total boiler surface, sq. ft.
S_r = radiant surface (equivalent), sq. ft.
C = coal burned per hour, lb.
H_c = B.t.u. in coal per pound
H_a = available heat per pound of coal
$H_a = H_c -$ loss (H, H_2O, CO, combustible in ash, etc.)
A = pounds of air per pound of coal.
Then $C/S_r = C_R$ = pounds of coal per square foot of radiant surface
$1/C_R = F$ = square feet of radiant surface per pound of coal burned
X = heat transmission per square foot of radiant surface per hour.

Hudson's formula reduced to

$$X = \frac{H_a}{F}\left(1 - \frac{A}{A+45F}\right)$$

This formula did not suit the known facts and was first modified by a term including the rate of running or the percentage of running with a modification of the constant. The formula then became

$$X = C_R H_a \left(1 - \frac{A}{A+42.5\sqrt{rF}}\right)$$

where the boiler rating r is a fraction.

Further investigation and more data enabled a further simplification and the following expression seems to fit all data fairly well:

$$X = C_R H_a \left(\frac{1}{1+\frac{A\sqrt{C_R}}{27}}\right)$$

The graph, Fig. 11, plotted on X and C_R as y shows the location of all the data which have been secured up to date.

The expression

$$\frac{1}{1+\frac{A\sqrt{C_R}}{27}}$$

is the fraction of the available heat in the furnace which is transmitted by radiation. The fraction

$$1 - \frac{1}{1+\frac{A\sqrt{C_R}}{27}}$$

is the fraction available for convection heat at the entrance to the convection surface and the temperature at this point must be:

$$t_e = \frac{H_a\left(1 - \frac{1}{1+\frac{A\sqrt{C_R}}{27}}\right)}{0.25(A+1)} + \text{temp. of draft} = \text{temp. entering convection surface.}$$

If the flue temperature t_f be known, then

$h = (t_e - t_f)0.25(A+1) =$ B.t.u. per lb. of coal absorbed by convection.

The B.t.u. per lb. of coal absorbed by radiation are

$$H_a\left(\frac{1}{1+\frac{A\sqrt{C_R}}{27}}\right) = H, \text{ and } \frac{H+h}{H_c} = \text{the boiler efficiency,}$$

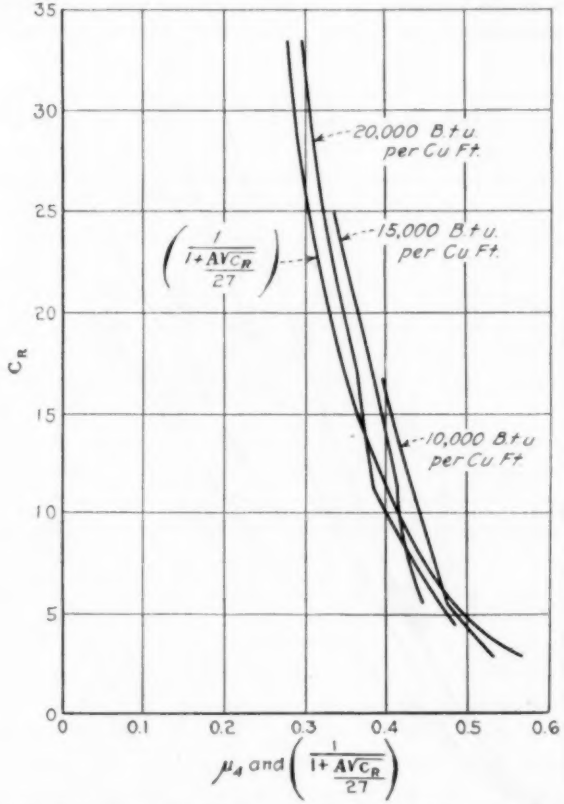

FIG. 10 COMPARISON OF WOHLENBERG'S COEFFICIENT μ_4 WITH THE HUDSON FORMULA MODIFIED BY ORROK

μ_4 = per cent of liberated B.t.u. absorbed by radiant surface.

$\frac{1}{1+\frac{A\sqrt{C_R}}{27}}$ = fraction of liberated B.t.u. absorbed by radiant surface.

thus giving a check which may be applied to boiler tests. This check is not as good as it should be since it is affected by after burning of CO and by leakage into the boiler beyond the furnace.

In the discussion of Wohlenberg's paper at the Milwaukee meeting (1925) the writer called attention to how closely that author's rational figure for μ_4 agreed with the writer's modification of

Hudson's empirical formula (see Fig. 10), but the most interesting check is obtained through well-conducted boiler tests where the "radiation and unaccounted for" percentage is positive and reasonably small. (See Fig. 11 and Table 6.) It has been tried on many of these tests from many sources with rather remarkable results as the variation is usually much less than the "radiation and unaccounted for" item. Some few examples are given in Table 6.

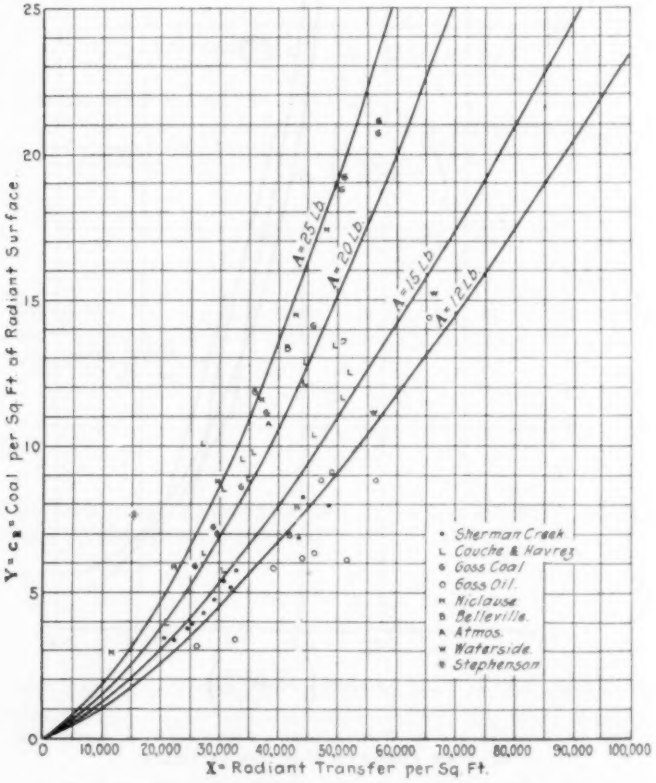

FIG. 11 RADIATION DATA AND CURVES FROM FORMULA

It should be observed that such convection heat as affects the radiant surfaces is considered as radiant heat, and such radiation from the gases as affects the convection surfaces is considered as convection heat. The convection temperature t_c in consequence may not be quite correct. However, the variations are not nearly as large as observed pyrometer variations, and the obtaining of accurate average furnace and flue temperatures is still an unsolved problem.

Hudson's curve is plotted with the unit transfer as abscissas and the total heat absorbed by the boiler per square foot of radiant surface as ordinates. Wohlenberg's μ_4 has the heat of the coal per square foot of radiant surface as ordinates while the fraction absorbed as radiant heat appears as abscissas. Broido uses the same coördinates. The writer has felt that on a plot such an

TABLE 6 SAMPLE CHECKS OF BOILER TEST RESULTS USING ORROK'S MODIFICATION OF HUDSON'S FORMULA

Station	Hell Gate		Lakeside		Conners Creek			Lansing
No. of test	1	2	22	12	64	62	60	4
Boiler	Springfield		Edgemoor			Stirling		Heine
Rating	170	228	167	241	59	142	196.4	294
C_R	3.66	5.07	6.36	9.87	1.92	4.67	6.74	10.1
$\sqrt{C_R}$	1.915	2.25	2.52	3.14	1.386	2.161	2.597	3.18
A	16.55	16.71	10.8	11.5	12.72	10.45	10.51	11.9
$\frac{1}{1+\frac{A\sqrt{C_R}}{27}}=\mu_4$	0.461	0.433	0.497	0.428	0.605	0.544	0.498	0.417
H_c	14261	14117	12544	12217	12549	12240	12200	13297
H_a	13480	13350	11880	11370	11621	11274	11042	12400
N	22800	29300	37600	48000	13500	28060	37050	52200
$0.25(A+1)$	4.39	4.18	2.95	3.125	3.43	2.86	2.88	3.23
$H=H_a\mu_4$	6220	5780	5900	4780	7030	6.135	5493	5170
H_a-H	7260	7570	5980	6560	4591	5139	5549	7230
$\frac{H_a-H}{0.35(A+1)}$	1650	1810	2026	2080	1338	1797	1927	2240
t_f	80	80	96	105	86	86	80	69
t_c	1730	1890	2122	2185	1424	1883	2007	2309
t_f	455	506	463	528	407	526	601	712
t_c-t_f	1275	1384	1659	1657	1017	1357	1406	1597
$h=(t_c-t_f)\times 0.25(A+1)$	5600	5780	4895	5180	3490	3881	4050	5150
$H+h$	11820	11560	10795	10050	10520	10016	9543	10330
Calc. eff., per cent.	82.9	81.7	86.1	82.2	83.8	81.9	78.2	77.7
Test eff., per cent.	83.2	80.4	86.2	82.3	80.9	81.7	78.5	77.2
Radiation and unaccounted for test	+0.09	+3.06	+0.5	+1.8	+2.1	+0.5	+0.5	+1.82

attempt at accuracy was not warranted and used C_R as ordinates with either $\mu_4 = \dfrac{1}{1+\dfrac{A\sqrt{C_R}}{27}}$ or the unit radiant absorption as abscissas. All of these diagrams are good, but the modified Hudson formula is so simple and so quickly worked on the slide rule that the diagrams are only good for comparison, a simple calculation giving all necessary quantities to check a complete boiler test or a new design.

In conclusion it must be stated that, whatever formulation or curve be used, the value of μ_4 or the fraction of heat utilized as radiant heat varies between 0.0 in waste-heat boilers to over 0.750 in certain experimental calorimeters, while the ordinary range lies between 0.250 and 0.600.

W. J. WOHLENBERG.[1] The author has brought out some very important facts, and designers of steam generators will do well to study carefully the whole of this paper. The matter pertaining to the radiating powers of gases at high temperatures included in Pars. 67 and 68 should be of particular interest and value to those engaged in the design and manufacture of the so-called boilers and economizers. It appears that by taking advantage of gaseous radiation a steam generator can be so proportioned as to yield a certain maximum efficiency with a certain minimum draft loss and for a relatively low cost. The results of Dr. Schack's research reproduced in the appendix are of great value in dealing quantitatively with the radiation problems involved in the design of such a steam generator.

The paper submitted at the last Spring Meeting by the writer and Mr. Morrow is mentioned several times, and on several counts these authors are put on the defensive by the present paper.

In the first instance, in Par. 23 occurs the statement: "Wohlenberg and Morrow have dealt with radiation from the smaller suspended particles of coal by *assuming* the radiation of the flame to be of minor importance." Let us see just what this assumption was. The author mentions the work of Callendar and Julius, as a result of which certain coefficients of radiation are obtained when the Stefan-Boltzman law is applied. He also reproduces the result of Dr. Schack's researches. When Dr. Schack's formulas are applied to the conditions investigated in the Wohlenberg and Morrow paper, practically the same coefficients of radiation result as those found by Callendar. It is interesting to note that Dr. Schack's work either assumes or substantiates the Stefan-Boltzman law. There is thus close agreement in results by either method. The coefficients which were used in evaluating radiation from the gaseous portions in the Wohlenberg and Morrow paper are those of Callendar. As a result of an application of such values, the gaseous portions were found to take a relatively small part in the total radiation. But however small this was, it was never neglected. The assumption to which the author refers is thus rather in the nature of a discovery, based on a logical procedure.

The question in the author's mind concerning the conclusion of Wohlenberg and Morrow on the relative magnitude of the gaseous radiation in the pulverized-fuel furnace probably arises from the fact that by experience we know that the radiant-energy absorption by cold surface in gas-fired furnaces is a considerable quantity. It should be noted, however, that radiated energy varies as, say, the fourth power of the absolute temperature, and only as the first power of the radiation coefficient. In the gas-fired furnace, therefore, a relatively small increase of the furnace temperature may, in a large measure, compensate for the low radiation coefficient. To

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illustrate this the graduate class in power at Yale University has applied the Wohlenberg-Morrow method in a cubical furnace of 6860 cu. ft. flame volume with a cold surface of 400 sq. ft. area at the top face. Three methods of combustion were investigated as follows:

1 Pulverized fuel, 20 per cent excess air
2 Coal burned on a grate at bottom of furnace, 20 per cent excess air
3 Same products of combustion as in (1) and (2), but flame was assumed to be a clear, non-luminous gas flame with no solid matter present.

For the third condition the radiation coefficients were assumed as those deduced by Callendar, which values agree with those used for the gaseous radiation in the Wohlenberg-Morrow paper. For an energy-liberation rate of 10,000 B.t.u. per cu. ft. per hr. the results given in Table 7 were obtained.

TABLE 7 FLAME AND REFRACTORY TEMPERATURES AND ENERGY-ABSORPTION EFFICIENCY UNDER VARIOUS CONDITIONS OF COMBUSTION

Fuel and Combustion Conditions	Flame Temperature T_U	Refractory Temperature T_R	Absorption Efficiency μ_4
(1)	2230	2000	0.40
(2)	2386	2025	0.36
(3)	2720	1353	0.25

It is to be noted that the coefficient μ_4 drops only from 0.40 to 0.25 when comparing pulverized-fuel conditions to the gaseous-combustion condition, but that the flame temperature in the latter case is about 500 deg. fahr. higher and the refractory temperature 650 deg. fahr. lower. This is exactly as it should be, because the low coefficient of radiation for the gases will increase the differential in temperature between both the flame and cold surface and the flame and refractory. This greater difference in temperature partially compensates for the low radiation coefficient, as a result of which the absorption efficiency μ_4 is higher than might at first be expected. Solid-fuel surfaces when present thus act as a means of discharging energy to colder surfaces at a higher rate than is the case for gases and thereby hold down the flame temperature. In so doing they reduce the flame-temperature magnitude to such a degree that the radiated energy from the gaseous portions is a relatively small part of the total radiation discharge. This temperature-damping influence of the black carbon surfaces is of course dependent in magnitude on the relative amounts of carbon surfaces to gaseous quantities as pointed out in the Wohlenberg-Morrow paper. In arriving at the particular magnitudes shown in Table 7 the Stefan and Boltzman law was assumed to apply for the burning gases as well as for the solid-carbon surfaces.

The use of the Stefan-Boltzman law for the radiation from the gaseous portions is of course not in accordance with the findings of Haslam, Lovell, and Hunneman, to which the author refers. Judging from the work of these experimenters, the radiation from the non-luminous gas flame may be in some way bound up with the reaction itself and does not obey the Stefan-Boltzman law. Whether there is a difference in the radiation laws applicable for the burning process and for the burned products, is still open to question. Probably there is, the latter obeying perhaps the Stefan-Boltzman law, and the former depending on additional factors. That such a difference should exist is reasonable when considering the source of radiation. Thus, according to Bohr's theory of the atom, a "quantum" of electromagnetic energy is radiated only when an electron travels from one quantum orbit to another. An atom supposedly contains electrons revolving in orbits about a nucleus, in much the same way as the earth and other heavenly bodies revolve about the sun. If the electrons remain in their respective concentric orbits, no energy is radiated or absorbed. Energy equilibrium is supposed to exist. The electrons have different amounts of energy in the different orbits. If, therefore, an electron is displaced between quantum orbits, energy is either radiated or absorbed, depending upon the direction of the displacement. Atoms and molecules which act as nearly perfect reflectors may be assumed, under such conditions, to have their electrons maintained in the several quantum orbits.

Now, during a combustion process there is undoubtedly a rearrangement of the electrons in the atom, whence radiation may occur, and the amount of this radiation may be subject to the influence of atom on atom. Perhaps some of the energy which would be radiated from the atom by itself is absorbed in providing the new electronic arrangement which results in forming the molecule. At any rate, energy is not radiated as rapidly as the Stefan-Boltzman law states that it should be if the results of Haslam, Lovell, and Hunneman are significant. It is thus apparent that before such questions can finally be answered it will be necessary to discover in more detail the arrangement of electrons and their variations in molecules. The engineer must therefore await further researches by the physicist before he is supplied with the correct laws of radiation from burning gases. Fortunately for our present theories, the energy involved in this part of the process is, as before shown, a small part of the total radiation when solid fuel is burned, thereby presenting nearly black radiating surfaces rather large in total extent.

This discussion leads directly to a second contention between the author and the writer. In Par. 31, the author expresses some doubt concerning the statements of Wohlenberg and Morrow on the influence of pulverization fineness on the radiating powers of the flame. Here again the findings are based on a logical procedure.

The coal weight in suspension within the flame volume at any instant presents the radiating surface. This weight is a function of the gas weight in the flame volume. If the coal quantity is divided into many small particles it will obviously present more black radiating surface than it would if divided into a comparatively few larger particles. The total radiation being proportional to the surface of the radiating body, it must increase with the fineness of pulverization. Thus the extent of the black radiating surface is increased under such conditions to such a degree as to

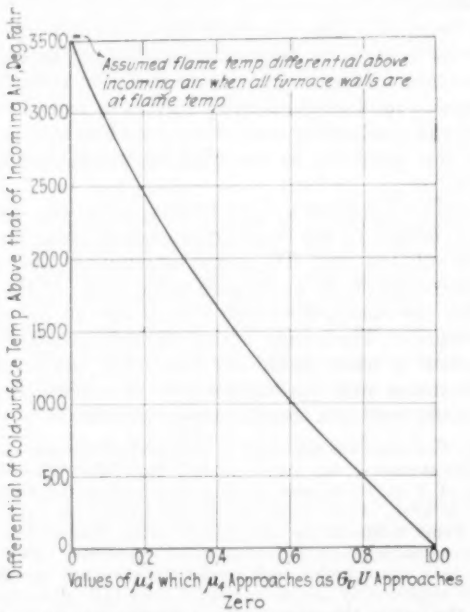

FIG. 12 VARIATION OF μ_4 WITH VARIATION OF TEMPERATURE DIFFERENTIAL

eventually overshadow the radiations emanating from the gaseous portions. Apparently, then, pulverization fineness is advantageous in permitting a higher speed of mixing, thus increasing the energy-liberation rate as well as that of the radiated energy, and thereby reducing the flame temperature to such an extent as to permit, for a given ash-fusion point, a higher energy-liberation rate without the formation of slag on the furnace walls.

In Par. 22 the author relegates certain treatments of the radiation problem to the academic cemetery. He then substitutes for this, in Fig. 3, a single curve between two coördinates, by means of which the value of μ_4 may presumably be found with sufficient accuracy for the designing engineer. He states that values of μ_4

taken from this curve are within 20 per cent of the correct value. Now it may be true that a value for μ_4 within 20 per cent is close enough for a designer's computations of overall efficiency. But as an indicator of the expected mean flame and refractory temperatures it seems that such an error is too large. Mean-flame-temperature values are, for a given fuel, indicative of the allowable limits of energy-liberation rate when considering furnace maintenance and operating difficulties resulting from slag formations. Let us therefore investigate more closely the limits of applicability of such a curve.

It will be shown that the author's choice for an ordinate is fortunate as to type, as for certain conditions the use of such a ratio eliminates furnace volume from radiation formulas. It does not, however, cancel out such other factors as variation in furnace symmetry and combustion conditions. As a first step in the examination of this curve let us see what its trends should be at the extremities.

If the ordinate is extended indefinitely, either the energy-liberation rate is infinite or the absorbing surface is zero. In either case μ_4 must be equal to zero. The upper extremity of the curve should thus become asymptotic to the zero value of μ_4. If the ordinate is zero, either the energy-liberation rate is zero or the cold-surface extent is infinite. The former is thus the only practical limit. The question then is, what should the value of μ_4 approach when the energy-liberation rate approaches zero in a finite furnace. The solution is obtained in a simple manner as follows:

CASE I. For convenience, take a cold-surface temperature equal to that of the incoming air, equal to 600 deg. fahr. absolute. Consider a furnace of 1 cu. ft. volume, with a fuel bed at the bottom equal to 1 sq. ft. in area. Find first the energy-liberation rate required to keep the flame temperature one degree above that of the cold-surface temperature, namely, at 601 deg. fahr. absolute. That combustion could not take place under such conditions does not in any way affect the results obtained by assuming that it does take place and the limits so set for the curves are correct. The radiation rate for this case is now

$$[(601)^4-(600)^4]1.62\times10^{-9}=17.5 \text{ B.t.u. per hr.}$$

Assuming 12 lb. of gas per lb. of fuel of specific heat 0.25, and fuel of a heating value of 12,000 B.t.u. per lb., and applying the equilibrium equation in the form

Rate of energy liberation = energy absorbed in furnace + energy absorbed by burned gases

the result is stated as

$$G_vU=17.5+0.25\times12\times G_v$$

whence

$$G_v=\frac{17.5}{11{,}996}=0.00146$$

But

$$\mu'_4=\frac{\text{Energy absorbed}}{\text{Energy liberated}}=\frac{17.5}{0.00146\times12{,}000}=0.999+$$

It is thus apparent that as G_vU approaches zero, μ_4 must approach unity for this case. If convection in the furnace were considered, it

would still approach unity as a limiting value of μ_4, the curve being, in this case, slightly lower all along. This discussion is merely to prove that it does approach unity as a limit under these conditions.

CASE II. If the cold surface absorbing heat is at a temperature above that of the incoming air, then obviously no heat can be absorbed by radiation or convection unless the furnace temperature is slightly above that of the surface. Let us assume a cold surface at 500 deg. fahr. = 960 deg. fahr. absolute. By applying the same method it is found that the limiting value of μ'_4 is 0.855+. If the surface temperature is 300 deg. fahr. the limiting value of $\mu'_4 = 0.92+$.

The complete range of limiting values of μ_4 for indefinitely small rates of energy liberation is shown in Fig. 12. It is apparent, for the usual range of cold surface temperatures, that the limiting value of μ'_4 or μ_4 must approach a value in the vicinity of 0.9.

Now let us see what is the trend of the author's curve in this respect. In Fig. 13 it is the curve *G*. This, extended, cuts the zero energy-liberation line at a value of $\mu_4 = 0.62$. Obviously, its trend at the lower end cannot be correct. On the same chart are shown curves plotted from the values of Wohlenberg and Morrow. These appear to have the proper trend. It is further noticed that at its upper extremity the author's curve closely approximates the (*A*) furnace of Wohlenberg and Morrow, which contains cold surface at the top face only. In the lower extremities it cuts across the several furnace curves. It may thus, even for these so-called theoretical furnaces, be considered as perhaps a mean of all of them. It appears, however, that a better representation of this is obtained by having a curve for each cold-surface fraction.

Let us examine next the variations of μ_4 with furnace volume for a given shape and surface arrangement; that is, for a given *furnace symmetry*. By equal furnace symmetry we understand furnaces in which equal solid angles are subtended between analogous parts; that is, from fuel surface to cold surface, fuel surface to refractory, refractory to cold surface, etc., as between large and small furnaces. For instance, all cubical furnaces having cold surfaces at the top faces, fuel beds covering the bottom faces, and refractories on the side walls, are symmetrical in this sense regardless of their size. If such a cubical furnace should have a volume of 1 cu. ft. and the energy-liberation rate is 10,000 B.t.u. per cu. ft. per hr., Broido's ordinate $\frac{G_U U}{S_\sigma}$ has a numerical value of 10,000. If the volume is 8000 cu. ft. the energy-liberation rate which will yield the same ordinate and hence the same chart value of μ_4 is $\frac{400}{8000} \times 10{,}000 = 500$ B.t.u. per cu. ft. per hr. Now let us see if under these two conditions the same value of μ_4 can be substantiated by theory.

In terms of the symbols used in the Wohlenberg-Morrow paper the expression for μ_4 may be stated in the form

$$\mu_4 = \left\{ (1-\beta_{CN}) \left[\frac{K_1 K_3}{(1-\beta_{RN})K_1 + K_3} + K_2 \right] (T_U^4 - T_C^4) + Z_C \right\} \div$$

$$\left\{ \Sigma G_B \Delta h_B + (1-\beta_{CN}) \left[\frac{K_1 K_3}{(1-\beta_{RN})K_1 + K_3} + K_2 \right] (T_U^4 - T_C^4) + Z_C \right\}$$

if the radiation loss from the furnace walls and heat taken up from the furnace by the incoming air is neglected. In this formula all

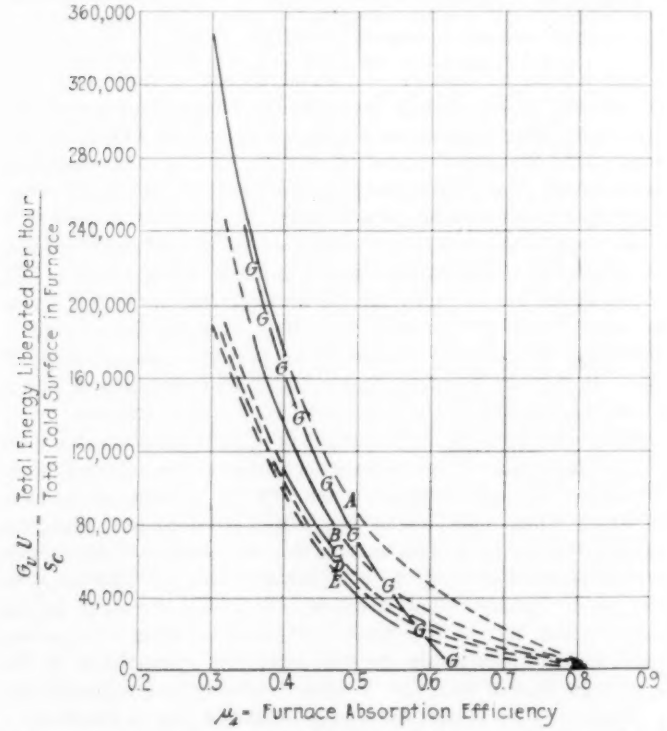

FIG. 13 COMPARISON OF BROIDO'S CURVE FOR VALUES OF μ_4 WITH THOSE OF WOHLENBERG AND MORROW

Curves *A*, *B*, *C*, *D*, *E* apply to cubical furnace for pulverized coal. Values of μ_4 taken from Wohlenberg and Morrow for corresponding values of Broido's ordinates $\frac{G_U U}{S_C}$.

A = ½ cold surface; *B* = ½ cold surface; *C* = ½ cold surface; *D* = ¾ cold surface; *E* = all cold surface; *G* = mean curve of Broido.

β's represent coefficients of reflection, G_B is the burned-gas-mixture weight, lb. per hr., Δh_B is the heat capacity of such gases between initial and flame temperatures, and K_1, K_2 and K_3 are each radia-

tion constants involving the coefficients, radiation angles, and surfaces involved, such as fuel, cold surface, and refractory. Z_C represents the heat absorbed by convection, and T_U and T_C are respectively the flame and cold-surface temperatures.

Suppose now that fuel is carried on a horizontal grate at the bottom of the furnace and that the combustion rate in pounds of coal per square foot of grate surface per hour is constant. The total energy-liberation rate then increases in proportion to the grate area if the furnace increases in volume. If furnace symmetry is maintained, all surfaces including the grate area vary as the two-thirds power of the furnace volume. Consequently, each of the quantities G_B, K_1, K_2 and K_3, and Z_C will be a product of some factors, one of which in each case is a surface. If furnace symmetry is maintained, corresponding surfaces as between large and small furnaces will, as stated, be proportional to

$$(\text{furnace volume})^{\frac{2}{3}}$$

Each of the above quantities may now be replaced by a product which has as one factor $(\text{furnace volume})^{\frac{2}{3}}$ and as a second factor that part of the quantity which with the first factor yields numerically the several values. Let us denote the second factor by the symbols R_B, R_1, R_2, R_3, and R_C. Thus $K_1 = R_1\ (\text{vol})^{\frac{2}{3}}$, $K_2 = R_2\ (\text{vol})^{\frac{2}{3}}$, ... etc.

Substituting, the result is

$$\mu = \left\{ (1-\beta_{CN}) \left[\frac{R_1 R_3 (\text{vol})^{\frac{4}{3}}}{(1-\beta_{RN}) R_1 (\text{vol})^{\frac{4}{3}} + R_3 (\text{vol})^{\frac{2}{3}}} + R_2 (\text{vol})^{\frac{2}{3}} \right] \right\} \div$$

$$\left\{ (\text{vol})^{\frac{2}{3}} \Sigma R_B \cdot \Delta h_B + (1-\beta_{CN}) \left[\frac{R_1 R_3 (\text{vol})^{\frac{4}{3}}}{(1-\beta_{RN}) R_1 (\text{vol})^{\frac{4}{3}} + R_3 (\text{vol})^{\frac{2}{3}}} + R_2 (\text{vol})^{\frac{2}{3}} \right] (T_U^4 - T_C^4) + R_C (\text{vol})^{\frac{2}{3}} \right\}$$

Inspection shows that the factor $(\text{vol})^{\frac{2}{3}}$ will cancel out of the latter expression. If now the flame temperature T_U can be proved independent of the size of the furnace when symmetry is maintained, for a given rate of energy liberation per unit surface, the absorption efficiency μ_4 itself must be independent of the furnace volume. We apply the equation of equilibrium which is exactly the divisor of the above expression placed equal to the total energy-liberation rate in B.t.u. per hr. This is equal to the product $G_U U$, in which G_U represents the pounds of fuel burned per hr. and U the calorific value or energy liberated per pound of fuel. Obviously, G_U is proportional to the grate area, whence we may substitute for it a product $R_U \times (\text{vol})^{\frac{2}{3}}$. Hence in the equilibrium equation the factors $(\text{vol})^{\frac{2}{3}}$ vanish, whence T_U is independent of the

furnace volume. We may now state a very important relation as follows:

> When a given solid fuel is burned with a given air excess on a bed or grate in a furnace so that energy is liberated at a constant rate per square foot of grate area, the flame and refractory temperatures and the absorption efficiencies are independent of the furnace volume if the furnace symmetry is not altered.

Now the Broido coördinate reduces the energy-liberation rate to that per square foot of cold surface. But if furnace symmetry is maintained, the cold surface, as between a large and a small furnace, is proportional to the grate area, whence the ordinate might just as well be taken as the energy liberated per hour per square foot of grate area, per square foot of horizontal cross-section, or even per square foot of refractory surface. The former, however, is more significant in that it expresses the combustion rate in terms of units which are already in common use. All such ordinates might otherwise be called ordinates of *constant symmetry*, as when the *furnace symmetry* remains constant for a given value of such ordinates the *radiation symmetry* is likewise constant, regardless of the size of furnace. In other words, the same values result for T_U, T_R, μ_4, etc. It must be noted, however, that these are deduced only for those furnaces in which coal is burned on a grate.

Let us now investigate the influence of variations in furnace volume on μ_4 for constant furnace symmetry other than solid-fuel surface when coal is burned in the pulverized form. As before, the solid angles depend only on furnace symmetry and are hence independent of volume. The radiating black fuel surface, now in suspension in the flame, varies in some manner with both fineness of pulverization and furnace volume as shown by Figs. 6 and 7 of the Wohlenberg-Morrow paper, but not as the two-thirds power of the volume. The refractory and cold surfaces, however, still vary as the two-thirds power of the furnace volume, whence not even furnace volume may be cancelled out of the expression for μ_4 for such cases. A single curve cannot therefore represent all furnaces of constant symmetry, and, where the volume varies greatly, distinct curves will be necessary for different volumes as well as for different conditions as to symmetry.

The foregoing discussion has shown, when a furnace is provided to burn coal on a grate, that the quantities T_U, T_R and μ_4 may be related by a single curve to any one of a group of the ratios

$$\frac{\text{total energy liberated per hour}}{\text{surface}}$$

if furnace symmetry is maintained throughout a range of furnace volumes and for given combustion conditions. The question now

is, what surface, included in the denominator, will yield the most useful chart? Returning for a moment to Fig. 13, it is noted that the author's choice of a denominator for this expression places the several curves in an order which is the reverse of that which would naturally be expected. Thus a larger cold-surface fraction results in a higher value of μ_4, other conditions being the same. In this

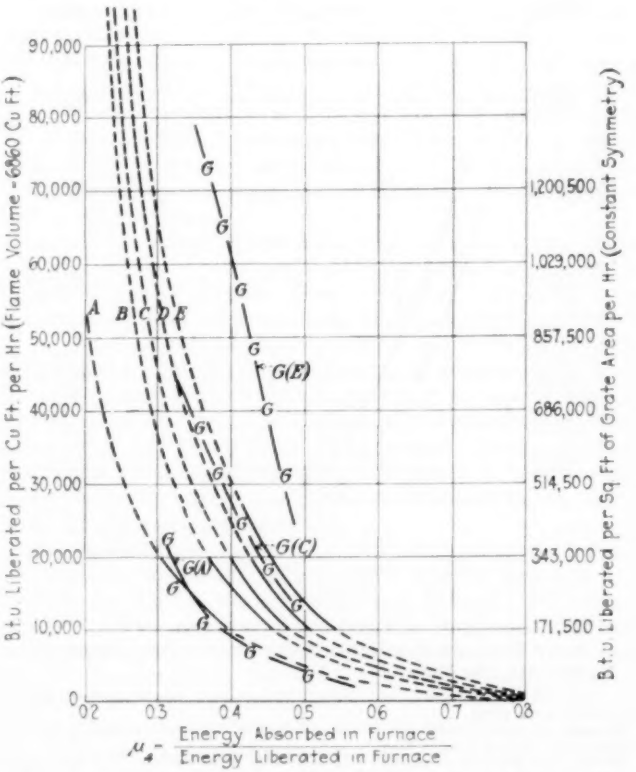

FIG. 14 CURVES $G(A)$, $G(C)$, $G(E)$ FOR FURNACES A, C, AND E IN WHICH VALUES ARE TAKEN FROM BROIDO CURVE G

Curves A, B, C, D, E apply to cubical furnace for pulverized coal. Values of μ_4 taken from Wohlenberg and Morrow.
A = $\frac{1}{4}$ cold surface; B = $\frac{1}{2}$ cold surface; C = $\frac{1}{3}$ cold surface; D = $\frac{2}{3}$ cold surface; E = all cold surface.

figure this increase is accomplished by a downward shift of sufficient magnitude, rather than by a direct right shift on the μ_4 scale when the cold-surface fraction increases.

If, however, the surface factor is taken as equal to the grate area or horizontal cross-sectional furnace area the curves appear in

their natural order as shown in Fig. 14. The constant-symmetry ordinate scale is shown as based on the grate area, but since this was taken as equal to the bottom face of the cubical furnace there shown, it might perhaps better have been labelled " B.t.u. per sq. ft. of horizontal cross-sectional furnace area per hr." The other ordinate scale being on the furnace volume base has numer-

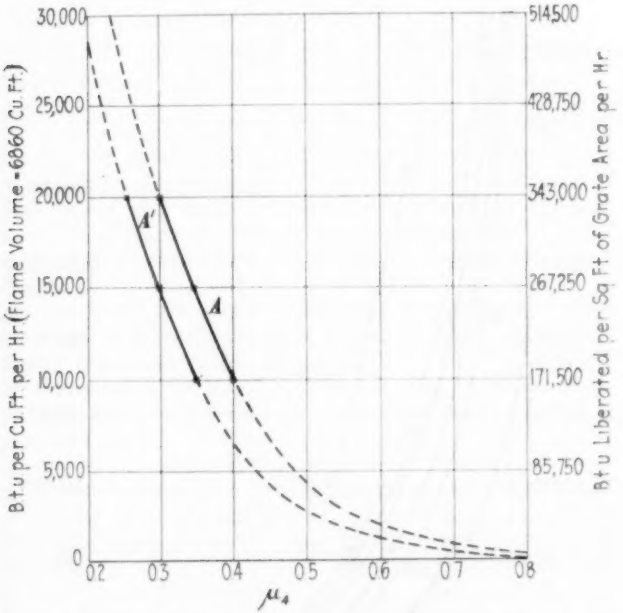

FIG. 15 COMPARISON OF VALUES OF μ_4, PULVERIZED FUEL VS. STOKER, IN SAME FURNACE AND UNDER SAME COMBUSTION CONDITIONS; I.E., EXCESS AIR AND FUEL

$A = \frac{1}{4}$ cold surface (pulverized coal); $A' = \frac{1}{4}$ cold surface (stoker, 400 sq. ft. projected grate area).

Furnace in each case cubical (20 ft. x 20 ft. x 20 ft.); cold surface at top face of cube; coal, Illinois (Saline Co.).

Moisture, 6 per cent; volatile, 32.4 per cent; fixed carbon, 54.3 per cent; ash, 7.3 per cent; B.t.u. as fired = 12,800; pulverized coal, 75 per cent through 200-mesh; 20 per cent excess air in both cases.

ical values which are applicable only to a flame of a given size, i.e., 6860 cu. ft., the volume always being such that the quotient

$$\frac{\text{(B.t.u. per cu. ft. per hr.)} \times \text{(flame volume)}}{\text{grate area}}$$

is equal to the value of the corresponding constant-symmetry ordinate opposite it. Since the values shown for furnaces A, B, C, D, and E are taken from Wohlenberg and Morrow as computed

for pulverized fuel they are not strictly applicable to stoker-fired furnaces.

On the same chart are shown values of Broido's (G) curve as applied to three of the furnaces. Obviously, the trend of the latter values for the completely black furnace (E) is not correct.

To indicate by comparison how much the values might be altered if the same amounts of energy were liberated under the same combustion conditions from fuel carried on a grate at the bottom face of the furnace, the Wohlenberg and Morrow method was applied, by some graduate students at Yale, to such a case with the results shown in curve A' of Fig. 15; the curve A, as before, representing similar conditions in the pulverized-fuel furnace. If the excess-air fraction had been taken higher, A' would of course have been shifted a somewhat greater distance to the left. For the furnaces with larger surface fractions the shift will of course be somewhat less.

In view of the foregoing discussion on furnace symmetry, curve A' can be taken as strictly applicable to a furnace of any size of the given symmetry with the given combustion conditions when the constant symmetry ordinate scale is used. Reference to curves B, C, D and E of Fig. 14 together with A and A' in Fig. 15 thus enables a fair estimate of the variations to be expected in μ_4 for a wide range of variations of furnace symmetry for the combustion conditions shown. Curve A' and approximated neighboring curves (not shown) should also hold for a wide range of furnace volumes. For pulverized fuel, however, a wide difference in volume from that for which the curves are shown undoubtedly should be the subject of a separate investigation.

In general, for furnaces provided to burn coal on a grate, distinct curves should be included for differences in furnace symmetry and combustion conditions. For pulverized coal distinct curves should be included for the above and also for considerable differences in furnace volume and perhaps pulverization fineness. Not until this whole field has been more thoroughly investigated can we assume any one or a few curves as sufficient.

In conclusion it may be said, although the Wohlenberg and Morrow method leads to consistent radiation values, the whole group of curves should perhaps be shifted somewhat either to the right or to the left. What experimental data the author includes appears to show that the shift should probably be to the right. However, these data are in themselves not consistent enough to indicate conclusively what the shift should be. Undoubtedly there were considerable errors as well as inconsistencies involved in the observations and perhaps in the interpretation of the observations.

A. G. CHRISTIE.[1] The author suggests that the cooling effect is probably confined to a narrow envelope along the side walls. The writer had the same opinion a year ago but is now less certain in view of further data on furnace phenomena. It is well known that glowing carbon particles radiate heat at a high rate to any exposed cold surfaces that can be "seen" by the particles. The author submits data which indicate that carbon dioxide and water vapor in the furnace gases also possess radiating properties. We do not know the effect of these gases on the radiating power of the carbon particles, and hence cannot fully appraise the effect of water-cooled walls on the furnace flame.

A substance may be transparent or opaque to radiant heat. Transparency to light does not always indicate that the material will be transparent also to radiant heat. Take the case of water which is transparent to light but quite opaque to radiant heat. When a substance is opaque to radiant heat it must absorb all such heat rays that reach it and in its turn radiate an equal amount of heat if its temperature is to remain constant, or all radiant heat may be reflected.

Our furnace problem resolves itself into a determination of the action of carbon dioxide and water vapor in the gases of combustion. If these intercept the heat rays from carbon particles or reflect them, it is possible that no cool surfaces can be seen by a carbon particle near the center of the furnace due to the interception of the heat waves by intervening gas masses of carbon dioxide and water vapor. Then only the outer envelope of the flame will radiate to the side walls. Further study must therefore be given to the radiant-heat effects of these two gases.

The author has suggested that the projected area of the tubes be assumed as the heat-absorbing surface. A more rational method would be to assume the whole exposed surface of the tubes as heating surface. In the case of the Murray furnace, this would consist of the exposed half of the tube and the exposed area of the fins.

In Par. 52 the author states that the furnace with large heat-storage capacity enables rapid increases of rate of driving. Such increase requires increased heat transfer, which can only result from higher temperature as a result of the combustion of more fuel. Increased rates of driving can be more easily obtained with water-cooled walls than with brick walls with large heat storage, for the water-cooled walls will respond more quickly to slight changes in furnace temperature and to increased heat radiation.

ANDREW A. BATO.[2] A new way to obtain further data concerning the radiation from the flame in the boiler furnace, together

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[2] Mechanical Designer, Public Service Production Co., Newark, N. J. Assoc-Mem. A.S.M.E.

with approximate results, is suggested in the following method, to be used in stoker-fired furnaces. It represents a trial to prove some facts indicated in the paper under discussion, and is meant as a suggestion for research work.

The difference between the "calculated" or "theoretical" and the "actual" furnace temperatures has been considered heretofore as one item, representing the total heat radiated by the fuel bed and the flames combined. This item can be divided into the radiation from the fuel bed and that from the flame by making a heat balance representing conditions, first for the surface of the fuel bed, and secondly for the top of the furnace, i.e., for the lowest plane of the water tubes.

For this end gas analyses, simultaneously with measurements of temperatures, are to be taken at the surface of the fuel bed and immediately below the lowest row of the water tubes at a sufficient number of points, together with the weighing and sampling of the coal fired and possibly with the determination of the heat content of the steam developed in the tubes surrounding the furnace, during a certain testing time. The gas analysis at the surface of the fuel bed will give a great content of unburned gases representing a certain heat value. If from the total heat value of the fuel burned we subtract the heat necessary to produce the actual surface temperature of the fuel bed plus the heat contained in the unburned gases, a remainder will be obtained representing the radiation from the surface of the fuel bed.

The gas analysis at the water tubes will give much less or no unburned combustible gases, with a small or no heating value. The temperatures will or will not show an increase from the fuel bed to the water tubes, or they will show even a decrease, depending on the amount of gases burned in the furnace and on the absorbing and cooling effect of the furnace walls.

The heat balance for the top of the furnace will indicate that the heat generated by the combustion of the unburned gases less the heat necessary to raise the temperature from that measured at the surface of the fuel bed to that measured at the lower edge of the tubes is equal to the heat radiated by the flames plus a small amount transmitted by conduction.

There are already sufficient data available to permit a rough estimate based on this method. They are contained in Technical Paper No. 137 of the U. S. Bureau of Mines,[1] giving gas analyses and temperatures at the surfaces of various fuel beds, not only of hand-fired furnaces, but also of an underfeed stoker, with not much difference in the amount of unburned gases obtained through the two methods of firing. This is explicable by the fact that the unburned gases consisted mainly not of volatile matter, but of carbon monoxide.

[1] Combustion in the Fuel Bed of Hand-Fired Furnaces, by Henry Kreisinger, F. K. Ovitz, and C. E. Augustine.

The data were obtained from tests conducted in a special round furnace 13½ in. in diameter, with firebrick only to absorb radiated heat. Coke, anthracite of 5.28 per cent volatile matter, and Pittsburgh coal of 34.27 per cent volatile matter were fired in three series of experiments, in fuel beds 6 in. and 12 in. thick. The rates of combustion varied from 20 to 130 lb. per sq. ft. per hr. The Jones underfeed stoker was tested from 75 to 130 per cent of rated capacity with New River coal.

The gas samples taken at the surface of the fuel beds contained amounts of unburned gases representing heat values up to 70 per cent of the heating value of the coal fired. Unfortunately the great majority of the tests showed a great lack of air, as indicated by the lack of oxygen in the gases. They thus are not directly applicable to conditions as found in actual practice. There are, however, a few results obtained with anthracite showing no lack of air, and giving about 32 per cent of the heating value of the fuel contained in unburned gases. Another publication of the Bureau of Mines, Bulletin 135,[1] gives results of tests conducted in a special furnace 5 ft. square, 30 ft. long, and fired with a Murphy stoker. It reports a case with the following data: Fuel, Illinois coal with a volatile content of 34 per cent; rate of combustion, 38 lb. per sq. ft. per hr.; 50 per cent excess air and 44 per cent of the heating value of the coal fired present in the unburned gases at a distance of 5 ft. from the gate. Further data will possibly be found in technical literature.

For the following example it will be assumed that the gases leaving the fuel bed, consisting of a high-volatile coal, contain 35 per cent of the heating value of the fuel in the form of unburned gases. Furthermore, it will be assumed for the sake of brevity that the actual and theoretical increases of temperature are directly proportional to the heat liberated. The calculated temperature of combustion will be assumed to be 3700 deg. fahr. As 35 per cent of the heating value of the fuel is contained in the gases, 65 per cent will be liberated in the fuel bed, and the calculated temperature at the surface of the latter will be 2440 deg. fahr. Instead of this, measurements give 1900 deg. fahr. as a result, corresponding to only 50 per cent of the heating value of the fuel, and leaving 15 per cent for the radiation from the surface of the fuel bed. This figure corresponds to the difference of the calculated and actual temperatures.

In the case of the cubical furnace with water-cooled walls this amount checks with the result calculated on the basis of the Stefan-Boltzman law, assuming that 30 lb. of 13,000-B.t.u. coal were fired per sq. ft. per hr., with an allowance for waste in the ashes. Thus the heat balance for the surface of the fuel bed will indicate that 50 per cent of the heat has been used to reach the

[1] Henry Kreisinger, C. E. Augustine, and F. K. Ovitz: Combustion of Coal and Design of Furnaces.

temperature of 1900 deg. fahr., 35 per cent is contained in the unburned gases, and 15 per cent has been radiated, giving altogether 100 per cent.

The gases will leave the fuel bed at a temperature of 1900 deg. fahr. and a heat content equal to 35 per cent of the heating value of the fuel in the form of unburned gases. The temperature of the gases will be about 2200 deg. fahr. on reaching the tubes, which will not necessarily be the highest temperature in the furnace. The rise from 1900 deg. fahr. to the temperature of 2200 deg. corresponds to 8 per cent of the heating value of the fuel. This, subtracted from the 35 per cent contained in the unburned gases, will leave 27 per cent for the radiation by the flame, provided the combustion of the gases has been completed in the furnace.

The method described above gives the possibility of checking and cross-checking the data obtained in every way.

As the surface of the fuel bed radiated 15 per cent of the total heating value of the fuel and the flame radiated 27 per cent, it follows that the radiation of the flame will be 180 per cent of that of the fuel bed for this particular case. This amount, reduced to the temperature of the fuel bed, will be only 140 per cent of the radiation of the fuel bed, and when divided by five for the five radiating surfaces of the flame it will give 28 per cent for an area equal to that of the fuel bed. It is interesting to note that the fuel bed of 1900 deg. fahr. sends heat rays through the flames having a higher temperature.

Tests carried out by Lent and Thomas, as given by Dr. Schack, and published in *Mechanical Engineering*,[1] may be quoted to show the probability of such higher values for the radiation from the flames of high-volatile coals. They found that the flame of blast-furnace gas made strongly illuminating by the addition of benzol gave a radiation approaching that of the black body, and about four times as high as without benzol, although the temperatures were kept at the same level. Wohlenberg and Morrow give values up to 40 per cent for μ_1, the ratio of the radiances of the pulverized-coal flame and the black body, depending also on the volume of the furnace. As the same figure for a clear and non-luminous flame amounts to 7 per cent and for a luminous gas flame amounts to 12 per cent, the value of 28 per cent as deduced above is acceptable for the flame of stoker-fired coal with much volatile matter, as ranking between the latter and the 40 per cent for pulverized-coal flame.

The results of temperature measurements at the side walls of a boiler furnace, as published in Table 1 show that the cooling effect of the side walls is still strongly perceptible at a distance of 4 ft. from the wall, as the temperature of the gases changes at a rate of 100 deg. fahr. per foot of distance between the third and fourth

[1] Vol. 47, no. 1, Jan. 1925, p. 41.

foot measured from the wall. Thus, considering also the probable great radiation of the flame of a high-volatile coal in a boiler of about 600 hp., having a furnace about 12 ft. wide, the whole mass of gases will be affected by the cooling effect of the side walls, and the smaller the furnace and the greater the content of volatile matter in the fuel, the greater will be the danger incomplete combustion. This is a fact well known from experience with internally fired boilers. When firing lignites having 30 per cent moisture and more, not only the completeness of combustion, but even the continuity of ignition may be in danger from the application of even the smallest surface of water-cooled side walls.

Finally it should be mentioned that, according to Roszak and Veron, hydrogen and hydrocarbons do not absorb and radiate heat to a perceptible amount.

E. H. TENNEY.[1] Some of the points touched on by the author, more particularly with respect to furnace temperatures and the relation of those temperatures to furnace-wall construction, have been the subject of much study and experimentation in both the Cahokia and Ashley Street stations of the Union Electric Light & Power Company of St. Louis. One of the main reasons why these matters are of such great importance to operators in the Middle West is the very low fusing temperature of the ash in the coal which they are compelled to burn.

The rates of combustion which are now possible with the use of powdered coal make it extremely easy to build up furnace temperatures higher than the fusing point of the ash. With such temperatures the suspended ash is in a molten condition, and unless chilled by coming in contact with walls sufficiently cool to reduce this temperature below the fusing point, there will be a rapid deposit of molten slag on the lower rows of tubes in the boiler. This, in itself, is serious and becomes subject to remedy when a sufficient volume of these high gases may be caused to come in contact with water-cooled furnace walls. If it were true that only a relatively thin envelope of the furnace flame is affected by wall conditions, then we could not expect relief from the deposit of slag on the bottom rows of boiler tubes in the case of our very low-grade western coals even with the water-cooled walls. It is quite apparent, therefore, that at high rates of combustion there must be sufficient turbulence given to the flame to insure contact of the maximum volume of gases with water-cooled surface. Such contact as this, in the case of air-cooled brick walls, would only result in lessening the time in which the brick would be destroyed. But in the case of the water-cooled surface, the possibility of relieving both the damage heretofore experienced to

[1] Chief Engineer of Power Plants, Union Elec. Light & Power Co., St. Louis, Mo. Mem. A.S.M.E.

refractories and the rapid slagging of boiler tubes is possible of accomplishment. We have so far been able to demonstrate the advantage to be gained with a turbulent flame on our Ashley Street furnaces equipped with partially refractory-lined water-cooled walls. At our Cahokia Station, with fin-tube water-cooled walls, we have made some improvement on the slagging of boiler tubes and have greatly relieved the maintenance costs incident to brickwork replacements.

The writer is more and more impressed with the fact that the problems which those designing and operating furnace equipment have to meet, are problems which are bound to vary with the characteristics of the fuel that is to be burned, and that the conditions to be met will in almost every case require individual treatment.

F. F. UEHLING.[1] Throughout the entire discussion of this most valuable contribution the writer has noticed that the term "flame temperature" has been very loosely used. What a thing is called makes little difference so long as all have the same understanding of it. However, flame temperature should not be confused with furnace temperature. Both terms are essential to the study of the important subject of radiation. They mean two decidedly different things. The author has illustrated this point exceptionally well by Fig. 2, in which curve *A* represents the mean flame temperature (temperature of combustion) and is entirely independent of the rate of combustion, while curves *B* and *C* represent mean furnace temperatures and are decidedly influenced by the rate of combustion, as the curves show.

The flame temperature is the temperature of the products of combustion, as caused by the chemical reaction between the combustible and the air at the instant the reaction occurs. This temperature therefore depends only on the kind of fuel used, the temperature of the entering air, and the analysis of the products of combustion. On the other hand, the mean furnace temperature is always lower than the mean temperature of the chemical reaction between the fuel and the air, and depends primarily upon how much heat is radiated in the furnace after the reaction between the fuel and the air has taken place.

The flame temperature, in the writer's opinion, should always be considered as the initial or maximum temperature of the products of combustion, and the term should never be used to represent furnace temperature. A differentiation between these two terms is of particular importance in a study of furnace radiation.

[1] President and Combustion Engineer, Uehling Instrument Co., Paterson, N. J. Mem. A.S.M.E.

H. H. SUPLEE.[1] Sometime between 1840 and 1850 the late Harrison Suplee, Jr., made a number of experiments on the transmission of heat through locomotive boiler tubes in connection with locomotive building in Russia. He divided the tubes by partitions into a number of spaces, filled with water, and measured the evaporation from each portion. The data are not immediately available, but it is known that he found the evaporation from the far ends of the tubes so meager that he changed his boiler design and made the first tubes shorter and the firebox longer. He went to Russia in 1840 and he returned in 1850, so that fixes the date approximately. That water-cooled walls are not new is evidenced by U. S. Patent No. 271,541, dated January 30, 1883, granted to the writer. This shows a complete set of water-cooled firebox walls. This combination was used very successfully prior to 1885.

E. G. BAILEY.[2] The author evidently favors the direct exposure of water-cooled tubes in furnaces as the remedy for the refractory trouble mentioned in first paragraph of his paper. The extent to which this water cooling is distributed over the different portions

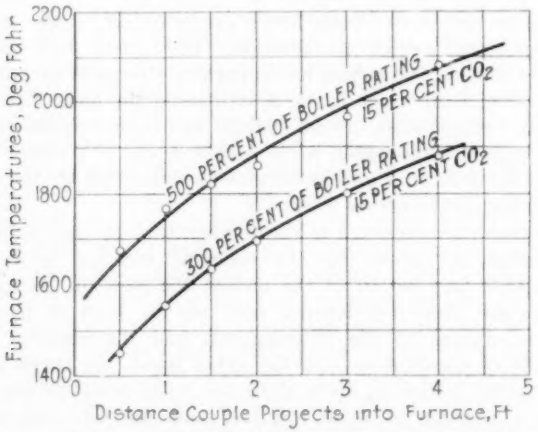

FIG. 16 PLOT OF AVERAGE TEMPERATURES GIVEN IN TABLE 1

of the furnace wall is summed up in Par. 53, the basis being the percentage of volatile matter, moisture, and ash in the coal being burned. This would seem to give lack of flexibility in changing from one coal to another throughout the life of the plant, and it would seem a better way to have a furnace that would withstand any reasonable variety of coals that might be available.

[1] University Club, Washington, D. C. Mem. A.S.M.E.
[2] President, Bailey Meter Company, Cleveland, Ohio. Mem. A.S.M.E.

If, as stated in Pars. 44 and 46, the furnace walls affect only the radiation of a relatively thin envelope of the furnace flame, then how can one cool wall materially prevent the rapid deterioration of an adjacent or opposite hot wall? Certainly the heat must radiate across at least half of the furnace before the cooling in this manner can be effectively carried out.

Fig. 16 is a plot of the average temperatures given in Table 1 of the various parts of the furnace shown in Fig. 1. From this it is noted that at 300 per cent of rating the furnace temperature is not above 1900 deg. fahr. at a distance of 4 ft. from the water-cooled wall, while at 500 per cent of rating the temperature does not come up to 2000 deg. fahr. until a distance of more than 3 ft. from the furnace wall is reached; yet in Par. 48 the author says that anything that will burn will ignite in the ordinary furnace temperature of about 2000 deg. fahr., intimating that below this temperature the ignition and completeness of combustion are problematical.

The influence which a cold furnace has on combustion is well illustrated every time a boiler is started up with cold walls. It is generally conceded that the hotter the furnace the better the combustion. The only limitation is to keep refractories below the fusing temperature of the ash coming in contact with them, and this is being satisfactorily done as the author notes in Par. 60.

THE AUTHOR. The author is indebted to Mr. Orrok for giving additional data or results of tests on heat absorbed by radiation in different steam-generating units. If additional points were added to the author's curve G in Fig. 3 corresponding to the tests given by Mr. Orrok, the slope of this curve would remain the same. It would be shifted only slightly to the left or given slightly lower values of μ_4. Points most deviating from the curve are those represented by the tests with the oil-fired locomotive. This is probably due to the fact that because of the long oil flame a considerable amount of it was absorbed in the firebox by convection.

The Hudson formula, modified by Mr. Orrok, is very simple and convenient in checking a complete boiler test. For new designs, however, where no definite data about the coal to be used or the method of operation are known, a curve showing the expected heat that will be absorbed by radiation for different conditions is more valuable.

The author fully appreciates the importance of fineness of the pulverization of coal. Fineness of coal undoubtedly affects the completeness of combustion. Very fine coal can be burned completely in a smaller combustion space. He does not, however, see how the fineness of pulverization can materially affect the radiation in a pulverized-fuel furnace.

It is a well-known fact that bituminous coal, while burning or when at high temperature, becomes rather sticky. There is no

doubt, therefore, that the small particles of coal combine together before they are completely burned, so that the results of any calculations in connection with the fineness of the coal are questionable. The total surface of all the particles of the coal, even if coarsely pulverized, is considerably larger than the total surface of the flame, so that the fineness of the pulverization can hardly have any effect on the heat radiated to the cool surfaces.

As explained by Professor Wohlenberg, theoretically curve G of Fig. 3 should be so shaped as to approach unity when the heat liberated per square foot of radiating surface approaches zero. Practically, however, this can never be the case. The fact alone that some heat is absorbed by radiation indicates that the gases leaving the furnace must be higher than the radiant heat-absorbing surface, so that not the whole heat liberated can be absorbed in the furnace. In a boiler furnace, the heat absorption by radiation very seldom exceeds 50 per cent of the total heat enumerated. Even with the newest design of steam generators, where the furnace is surrounded on all six sides with cooled surfaces, the heat absorbed by radiation never exceeds 60 per cent, so that curve G showing a maximum for μ_4 of 0.62 cannot be considered incorrect for practical purposes.

Professor Wohlenberg's theoretical discussion leading to the conclusion that separate curves are required for differences in furnace symmetry and combustion conditions is very interesting. It should be noted, however, that the maximum difference between the curves E and G(E), Fig. 14, at 50,000 B.t.u. liberated per cubic foot per hour is less than 20 per cent, which shows that the author's contention that his curve G is sufficiently accurate for practical purposes and can be used as a standard on which to base tentative calculations, has not been disproved by Mr. Wohlenberg's elucidation. It rather shows that even taking into consideration many variables, the difference is not sufficiently large to exclude the use of one curve. The furnace volume has unquestionable effect upon the heat absorbed by radiation for the same rate of heat liberation. For practical boilers, say, between 400 and 2000 hp., the difference will not be more than 20 per cent. When more data are available showing the influence of the various factors upon the heat absorption in boiler furnaces, we shall be able to compute separate curves for various conditions.

Mr. Bato's method of determining the radiation from the flue gases in the boiler furnace is very interesting. It depends upon accurate measurement of temperatures near the fuel bed and immediately below the lowest rows of boiler tubes. The difficulties in measuring gas temperatures in the furnace, however, is a known fact.

Mr. Bailey assumes that the author favors the direct exposure of water-cooled tubes in the furnace. In Par. 53 it is explained that the amount of water-cooled tubes exposed to direct radiation in a

furnace depends upon many conditions, the chief of which is the kind of fuel used. For certain fuels and under favorable conditions, the furnace can be covered completely with water-cooled tubes and complete combustion can be obtained. Where the fuel contains a high percentage of moisture and ash, it may be advisable to cover the water tubes partially or wholly with cast-iron blocks or with refractory.

Mr. Bailey wonders how cooled walls can materially prevent the rapid deterioration of an opposite brick wall if the furnace walls reduce the temperature of the gases only a few feet away.

As stated in Par. 25 of the paper, the radiation from the flame increases with each thickness up to about 24 in. Above that, no increase in radiation takes place, which means that a flame much over 24 in. thick is not transparent for its own radiation, which explains why only a comparatively thin envelope of the furnace flame is affected by the cold wall. At the same time, however, the flame may be transparent to radiation from one solid body to the other, between two opposite furnace walls.

There are two distinct purposes in applying water-cooled walls to steam boilers:

(1) To decrease the temperature in the furnace and by that, reduce the maintenance cost of the brickwork

(2) To absorb as much heat as possible by radiation and increase the capacity of the boiler.

In order to reduce materially the maintenance cost, it is only necessary to decrease the gas temperature in the furnace below the fusing point of the ashes. In most cases this can be accomplished with bare water-cooled tubes without affecting the combustion of the furnace, in which case there is no necessity for covering these tubes with cast-iron or refractories, which materially reduce the heat-absorption capacity of these tubes.

No. 1991

THE HEAT-BALANCE METHOD OF TESTING CENTRIFUGAL COMPRESSORS

By M. G. Robinson,[1] Lynn, Mass.
Associate-Member of the Society

Centrifugal compressors are often driven by steam turbines with and without intervening gear trains. To determine the shaft power transmitted to the compressor by its driver, from efficiency tests on the turbine and estimates of the gear losses, involves considerable labor, time, and expense. The Heat-Balance Method herein described is simpler, and more direct. In this method, the principle is used that a balance exists between the mechanical energy supplied by the rotation of the shaft and the heat-energy increase in the air which is being compressed, the heat-energy increase in the cooling water used, and the heat lost from the casing, bearings and packings. With proper testing apparatus and due precautions, this heat energy may be accurately measured, and hence the power input established. Very satisfactory results have been secured by the application of this method. There are given details of the fundamental theory, some representative test data, and a complete description of the apparatus used.

THE fundamental requisite of progress in the development of any apparatus is definite knowledge of performance. The designer must know from a thorough and reliable test what he has accomplished before he is ready with the next step. The builder of centrifugal compressors, because of the high rotational speed involved, employs steam turbines, and at times gear trains to drive his apparatus. When the efficiency of the driver under test conditions is known, the input to the compressor (which is represented, of course, by the output of the driver) can be computed, and hence the efficiency of the compressor established. However, the labor involved in making a test on the gear losses, or a steam rate power-output test on the turbine is very appre-

[1] General Electric Co., River Works.

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ciable, and often prohibitive. Sometimes, it is even impossible to reproduce on test the rated conditions of the driver, as is often the case with very high steam pressures or superheats; or, again, in connection with compressors designed to handle gas of low specific gravity. In the latter case, the machine when tested with air requires power in great excess over the rated, and hence the driver is of necessity operated under abnormal conditions. The thermodynamic transformations involved in interpreting the performance of the compressor with air to an equivalent performance with gas are usually simple, and can be accurately made. However, the efficiency of the compressor, or its probable power consumption, may be seriously in error, if based on a calculated output of the driver, since considerable extrapolation from known test data is often necessary on account of the test operating conditions being so far removed from the design conditions. The advantage of having some direct way, independent of the driver, of determining the power input to the compressor is therefore quite evident. This can be done by measuring the equivalent heat energy appearing during the process of compression. This principle has been made use of, to some extent, abroad, particularly in Germany. However, so far as the author is aware, there is no clear complete exposition of the subject available. The test reports, meager as they are, contain, as a rule, heat losses listed as "assumed"; the procedure used in the temperature measurements is almost never described; the basic factors adopted are seldom if ever stated, and their validity practically never discussed.

2 Within the last three years a great deal of testing has been done at the Lynn Works of the General Electric Co., by such a method, which may be called the Heat-Balance Method, with results fully trustworthy, having been again and again checked by independent means. The objection which has often been raised to any calorific method as showing too low an input has been met by the adoption, after thorough study, of proper values for the basic factors involved, such as c_p for moist air, radiation-loss coefficient, etc., as well as general refinements in the testing procedure. A set-up has been developed which is easy of manipulation, quick, inexpensive, and when due precautions are exercised, quite accurate.

Theory

3 The theory underlying the method is simple. It is based on the first law of thermodynamics that "Heat and Mechanical Energy are Indestructible and Interconvertible." If one is skilful enough to determine all the heat energy appearing during a mechanical action, this energy will "balance" the energy supplied by the mechanical action. Let us examine what happens during the operating cycle of a centrifugal compressor, and see how the principle stated could be applied. The driver, be it a turbine or a

motor, delivers a certain amount of mechanical energy to the compressor rotor. This energy is used up as follows:

a A part goes to increase the energy contents of the air which is being compressed and delivered into the discharge pipes. It manifests itself in a temperature and pressure rise and delivery of the air.

b A part is absorbed by the cooling water in the water jacket surrounding the air passages, and manifests itself in a temperature rise of the water.

c Some is dissipated into the surrounding atmosphere from the surface of the casing by conduction, radiation and convection.

d Some is lost in bearing friction, the heat developed being carried off by the oil, or else dissipated into the surrounding atmosphere.

e Some is carried off into the surrounding atmosphere by the air escaping through the clearance between the shaft and the packing on the high pressure end of the machine.

4 The problem which confronts one is then the evaluation of each of the items enumerated above. Usually, items *a* and *b* are of major importance. Item *a*, the energy absorbed by the air, depends upon the specific heat, the flow, and the temperature rise of the air. It can therefore be evaluated with a high degree of precision. In the case of compressors with no water cooling this energy represents practically the entire shaft input (see Appendix No. 2, Tables 1, 2 and 3). In water-cooled compressors, this energy may be only 50 per cent of the total in the case of low pressure machines, and even as low as 20 per cent in the case of high pressure multi-stage machines (see Appendix No. 2, Tables 4 and 5).

5 Item *b*, the energy carried off by the cooling water, depends on the weight of water used and on its temperature rise. It also can be determined accurately. In case of high-pressure multi-stage compressors this item is the most important one (see Table 5).

6 Items *c*, *d* and *e* in the average case amount to from 0.5 to 2 per cent each, near and above full load, and hence their magnitude does not have to be established with any great degree of precision, a 10 per cent error in estimating any of these items having an effect of less than $\frac{1}{4}$ per cent on the overall input. However, at partial loads the relative importance of items *c*, *d* and *e* becomes more marked. It follows that the results obtained by this method are apt to be in error at partial loads, since the nature of items *c*, *d* and *e* is such that it is difficult to establish them very accurately. (It may be noted in passing that the calculated outputs of most drivers are also subject to a greater degree of uncertainty at light loads.)

7 The test data given in Appendix No. 2, Tables 1 to 5, illustrate the actual energy distribution in a few representative cases, and so indicate the degree of refinement required in the determina-

tion of each item. These tests were carried out with such care that the summation of the items enumerated (a to e) actually shows the balance between the heat energy measured, and the known input of the driver. The usual agreement is within 2 per cent or better.

Theoretical Formulas and Basic Factors

8 The detailed formulas and other information required in the computation work are given fully in Appendix No. 1. For convenience in reference, in the general discussion which follows, the five fundamental expressions for the energy subdivisions, a to e, described in the article are also listed below:

$$\text{Air hp.} = Jc_p(T_d - T_i)w/550 \quad [a]$$

$$\text{Water hp.} = J(t_d - t_1)W/550 \quad [b]$$

$$\text{Radiation hp.} = JS(K/3600)(t_s - t_a)/550 \quad [c]$$

$$\text{Bearing hp.} = \text{known by test, or estimated from appropriate formula} \quad [d]$$

$$\text{Packing hp.} = Jc_p(t_p - t_i)w_p/550 \quad [e]$$

where:

J = mechanical equivalent of heat
c_p = specific heat of air at constant pressure
$T_d - T_i$ = temperature rise of the air
w = weight of air compressed
$t_d - t_1$ = temperature rise of the cooling water
W = weight of cooling water
S = exposed surface of compressor casing
K = coefficient of heat emissivity from casing
$t_s - t_a$ = temperature difference between surface casing and surroundings
$t_p - t_i$ = temperature difference between air leaking by packings and inlet air
w_p = weight of air lost through packing leakage.

9 The total power input is, of course, given by the sum, $a+b+c+d+e$. The similarity and the simplicity of the formulas, for both the air hp. and packing hp. is due to the well known fact that, since air may be considered a perfect gas, any energy increase in it is manifested by a temperature rise only. In the actual calculations care must be exercised in selecting the proper value for c_p as specified in Appendix No. 1, taking due account of its variation with temperature, pressure and humidity, as indicated in Figs. 1 and 2, in the light of the most recent experiments. Failure to do so may, in an extreme case, cause an error of as much as 4 per cent.

10 So much vagueness had usually surrounded item c, the energy lost by the casing through radiation, conduction and convection, that a special study was made of the subject. As is evident from the formula, c, three factors affect this item: (1) the surface

exposed (S); (2) the temperature difference between the surface and the surrounding atmosphere ($t_s - t_a$); and (3) the coefficient of emissivity (K).

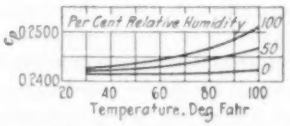

FIG. 1 VARIATION OF SPECIFIC HEAT OF AIR, c_p, WITH TEMPERATURE AND HUMIDITY

(Values for dry air based on tests by W. F. G. Swann.)

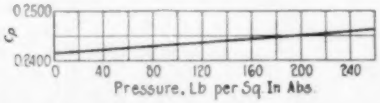

FIG. 2 VARIATION OF SPECIFIC HEAT OF AIR, c_p, WITH PRESSURE

(Values for dry air from tests by Holborn and Jacob.)

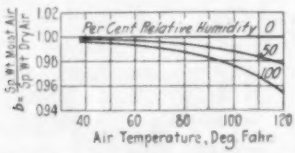

FIG. 3 EFFECT OF HUMIDITY ON SPECIFIC WEIGHT OF ATMOSPHERIC AIR

(Computations based on Marks' and Davis' Steam Tables.)

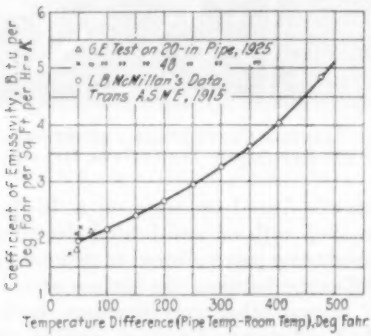

FIG. 4 EMISSIVITY TEST ON BARE PIPE

11 During a test, the surface is conveniently subdivided into portions of approximately equal temperature, and that temperature observed. The surrounding temperature is measured in a sufficient number of places to insure a representative value. The

total power is the summation of losses from the several areas. A proper value of the coefficient of emissivity may be selected from curve in Fig. 4 which is based on tests made by L. B. McMillan.[1] This coefficient gives the combined effect of heat loss by conduction through the surface, radiation from the surface and convection. Of these, the last named is of major importance. Although there appears to exist an appreciable variation in the coefficient with the temperature difference, it should be noted that as applying to compressor tests, values 2.0 to 3.0 will practically cover the range.

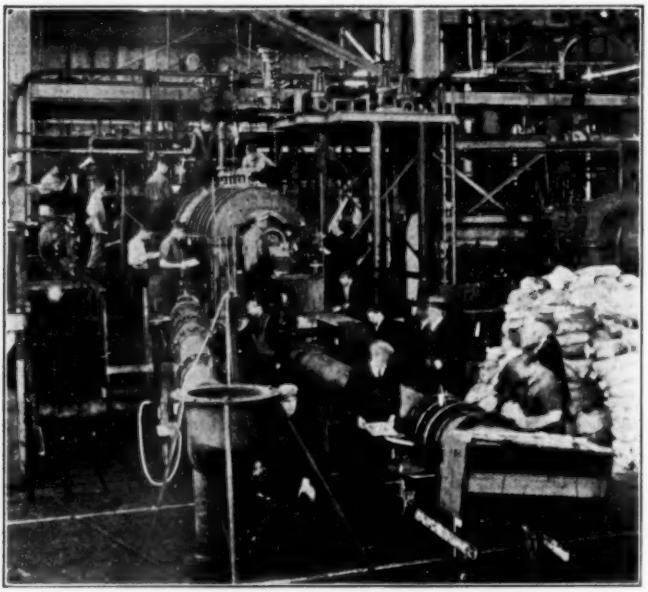

FIG. 5 GENERAL ELECTRIC CENTRIFUGAL COMPRESSOR. TURBINE-DRIVEN UNIT. SET-UP FOR EFFICIENCY TEST BY THE HEAT-BALANCE METHOD

McMillan's curve is based on tests on a 16-ft. length of 5-in. bare steel pipe filled with oil and heated electrically, the amount of energy supplied being just sufficient to maintain a given constant temperature difference between the surface and the surroundings. The results of several other experimenters, both by steam-condensation method and electrical method, were examined, and values of the same order of magnitude found.

12 As additional evidence, primarily to note the effect, if any, of diameter variations, a check was made by the writer personally,

[1] "The Heat-Insulating Properties of Commercial Steam-Pipe Coverings." Trans. A.S.M.E., 1915, vol. 37.

using the following method devised during tests on a big multistage compressor, similar to the one shown in Fig. 5. The discharge piping was assembled, using (1) a length of 20-in. pipe having an exposed surface of 211 sq. ft., and (2) a length of 48-in. pipe having a surface of 285 sq. ft. The entire air quantity handled by the compressor flowed through these pipes in series with a calibrated nozzle. The weight of air was found from the pressure drop across the nozzle in the standard manner. The surface of the pipes was, of course, at a temperature higher than that of the surroundings, and there was a definite loss of energy which manifested itself by a temperature gradient in the air flowing through the discharge pipe. A careful determination was made of the average air temperature at the beginning and end of both the

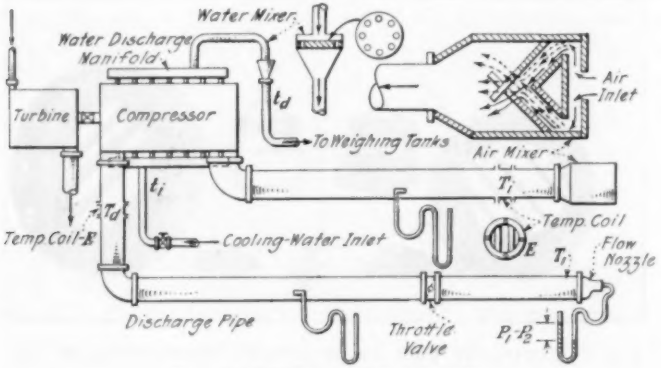

FIG. 6 DIAGRAM OF TEST SET-UP

20-in. and the 48-in. pipe, using both mercury and resistance thermometers.

13 Knowing the weight of air flowing through the pipe, the heat loss coefficient was found from the following relation:

$$3600 \times c_p(t_2 - t_1)w = KS(t_s - t_a)$$

where $c_p = 0.242$; $t_2 - t_1$ = temperature drop of the air between the two sections; w = weight of air flowing, lb. per sec.; S = surface exposed, sq. ft.; $(t_s - t_a)$ = temperature difference between the surface and the surroundings; and K the coefficient of emissivity.

14 The results of this test are shown on Fig. 4, and are quite consistent with McMillan's. This is especially encouraging when we consider the entirely different method adopted as well as the fact that McMillan's work was made on a laboratory scale while the test described was carried out on a large scale under actual test conditions on a big compressor. A point which is also worth noting is that in all other experiments there was no flow of the

fluid inside the apparatus. There is no reason to believe that flow would have any effect on K, so long as the *outside* surface temperature is measured. However, it was quite gratifying to actually note this from a test with varying velocities of flow.

15 The remaining formulas in the list do not require any further explanation beyond that given in Appendix No. 1.

Description of Apparatus and Procedure

16 The degree of accuracy which can be obtained from a heat balance test depends, wholly, on proper apparatus and the exercise of sufficient care in conducting the test. The general arrangement of the set-up used is illustrated by Fig. 5, and the diagrammatic sketch, Fig. 6. It will be most convenient to discuss the various features in order in which they affect the five subdivisions of the total energy input.

Fig. 7 Equipment Used for Measuring Temperatures by the Resistance Method

17 (*a*) *Power Absorbed by the Air.* As indicated elsewhere, this depends on the specific heat (c_p), the temperature rise of the air, and the flow. The variation of c_p with the moisture contents is shown on Fig. 1. During the tests, the relative humidity is determined from readings of wet- and dry-bulb thermometers, the sling-type instrument being preferred. The flow is usually measured by the nozzle method, as described in detail by S. A. Moss.[1] Care is taken to have a sufficient length of piping between any irregularity (such as an elbow or a throttle valve) and the nozzle to insure smooth flow to the nozzle. With proper piping and using nozzles (of the shape shown in the paper referred to) of 4-in. diameter and over, a discharge coefficient of 0.99 may be expected.

[1] The Impact Tube, Trans. A.S.M.E., 1916.

18 The determination of the true temperature of both the inlet and discharge air is probably the most difficult task in the method. When testing is done in a factory building where steam may be discharging in one corner, and cold outside air flowing some hundred feet farther, appreciable temperature variations are apt to exist, and convection currents are set up. A thermometer placed indiscriminately somewhere near the machine may differ several degrees from the true temperature of the air actually passing through the intake of the compressor. A thermometer inserted somewhere in the discharge from the compressor may be subject to the action of a stray air jet, and will not necessarily indicate the true average temperature of the discharge air.

19 To obviate these difficulties, there has been developed a system of resistance thermometers which are shown on Fig. 7. The basic principle is that of determining the temperature from a

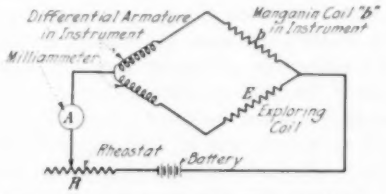

FIG. 8 DIAGRAM OF RESISTANCE-THERMOMETER CIRCUIT

change in the resistance of a known element. The resistance elements are usually inserted in a pipe section (of same diameter as the main piping) about 10 inches long and provided with flanges on each end. They are placed directly in the inlet and discharge pipes leading to and from the compressor. In this way, all the air is made to pass over the resistance elements, and a good representative average temperature is secured. The resistance wires are of copper, and are wound on four bakelite struts of streamline section, and connected in series. There are provided four tapped holes which enable mercury thermometers to be used in conjunction with the coils, and thus afford an immediate check on tightness of electric connections, grounds, etc.

20 The electrical circuit can be traced from Fig. 8. The indicating instrument is of the constant-current type having three scales covering a range of temperatures from 0 to 320 deg. fahr. In the operation of the device, the current in the circuit is held constant by means of the rheostat (R) at the design value of the instrument. The various resistances of the instrument are adjusted so that at 25 deg. cent. (77 deg. fahr.) the current arriving at the juncture of the differential armature through the manganin coil is equal to that arriving at the same junction through the exploring coil (E); this determines the normal point (zero deflection)

of the instrument. When the exploring coil (*E*) is at a higher temperature than 77 deg. fahr. its resistance is increased; the two sides of the circuit through the differential armature are unbalanced, consequently more current flows through coil (*b*) than through coil (*E*) and the pointer of the instrument moves up scale. In the case that the temperature of coil (*E*) is lower, coil (*b*) takes less current than coil (*E*) and the pointer moves down scale. The heating effect of the current supplied is negligible since it is small, and flows but for a few seconds at a time, i.e., while an observation is being taken. Frequent checks are made upon the constancy of both the resistance elements and the scale calibrations of the indicating instruments.

21 As an additional precaution there has been used of late an "air mixer" which can be seen on the end of the inlet pipe on Fig. 5 and in diagram of Fig. 6. In this arrangement the entire air quantity is taken through a horizontal opening of a fairly large wooden box so as to reduce the initial temperature variations. The air jets are then divided and quite thoroughly mixed, as shown on the diagram. By using this air mixer, very uniform temperatures across the section of the inlet are obtained.

22 (*b*) *Energy Absorbed by the Cooling Water.* This energy may be as much as 50 per cent or even more of the total energy supplied, and hence all precautions possible must be exercised in its determination. Fortunately, this is usually not very difficult. It depends on the product of the temperature rise and the weight of the cooling water which is being circulated. Usually there is practically no variation in the temperature of the water as it flows through the inlet pipe. Hence in determining its temperature all that is necessary is the use of a sufficient number of properly calibrated thermometers with sufficiently fine graduations. In all of the tests, here described, at least two, and usually three, thermometers were used. The problem is more troublesome on the discharge end, where an individual determination may not be truly representative. This uncertainty was overcome by using a "water mixer" which may be seen installed in the vertical section of the pipe in Fig. 6. The action of the mixer will be evident from the insert on Fig. 6. By having the main stream impinge upon the central solid portion of the plate the velocity of any individual jet if present is destroyed and a certain amount of mixing takes place. The water is then forced out through a number of small holes drilled near the outer periphery. The many jets discharging through these holes into the gradually converging cone are thoroughly mixed and when the stream issues from the cone the water has a uniform temperature throughout the section. A single thermometer properly graduated and calibrated gives a reliable and truly representative indication of the average temperature.

23 An attempt was also made at an earlier date to obtain the average temperature by using the resistance-thermometer method

similar to that employed on the air end. However, so much difficulty was experienced due to electrical leakage and temperature time lag that the method was for the time being abandoned and the water mixer adopted. During the entire test, the flow of the cooling water is kept constant by means of suitable hand-controlled throttle valves. The actual amount is determined usually by direct weighing in tanks specially set up for the purpose.

24 (*c*) *Energy Dissipated into the Surroundings from the Casing Surface.* During the progress of the test, careful measurements are made of the temperature of the various parts of the casing. The temperature of the surroundings is also observed. Each test point is held a sufficiently long time to insure constant thermal conditions having been attained. The coefficient of heat loss, K, is taken from the curve on Fig. 4, or at times determined by actual test on a pipe section simultaneously with the performance test proper using the method previously outlined. It should be noted that the energy loss from this source in case of water-cooled machines is small. In case of machines without water cooling, it may be of importance (see Appendix No. 2, Tables 1 to 5).

25 (*d*) *Bearing Hp.* Whenever possible, the bearing loss is determined by test. During the performance test, care is taken to have oil of same viscosity as that used during the bearing tests. When a calorific test is made, the oil and water should be weighed preferably to being metered through an orifice, and due precautions taken to run the test long enough to insure constant conditions.

26 (*e*) *Packing Hp.* The clearances between the packings and the shaft are carefully measured both during the process of assembly, and after the test. Throughout the test, observations are made as to the condition of the packings, so that any imperfection resulting in an unduly large leakage is quickly noticed.

Representative Test Results

27 Several attempts were made during the progress of the work to check the heat balance method by direct comparison of the power calculated from the measured input to a driver of known efficiency with the power obtained by the heat balance method. Motor-driven as well as turbine-driven sets, with and without an intervening gear train, were tried. The results reported here (see Appendix No. 2, Tables 1 to 5) represent tests, corresponding to conditions at which the output of the driver could be ascertained with a good degree of approximation (1 to 2 per cent). The losses of induction motors were determined from tests in accordance with the specifications of the A.I.E.E. standards of 1922. The outputs of the turbines were calculated from data on water brake and generator tests of similar units.

28 Table 1 applies to a 75-hp. motor driven set of the general construction shown on Fig. 9. It is a single-stage fabricated-steel

compressor, without water jacket and with the impeller wheel overhung from the motor shaft. It will be noted that the results show an agreement between the power input to the compressor as computed from the heat-balance method and that obtained from the measured input and the known motor losses within $2\frac{1}{2}$ per cent or better. The bearing losses are not included, being charged to the motor. The packing loss was negligible. The major item is, of course, the air hp., as in any unit without water cooling.

29 Table 2 applies to a 175-hp. motor driven set, the compressor being coupled to the motor through the medium of a gear train (see Fig. 10). The efficiency of the motor was known to a

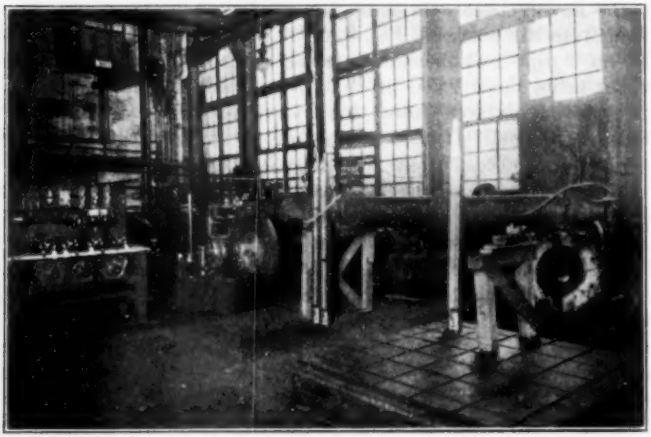

FIG. 9 GENERAL ELECTRIC CENTRIFUGAL COMPRESSOR MOTOR-DRIVEN SET. SET-UP FOR EFFICIENCY TESTS

fair approximation ($\frac{1}{2}$ to 1 per cent). The losses of the gear train were determined by a separate test, the gear being uncoupled from the main motor and driven idle by means of a small calibrated motor. The tabulation shows extremely satisfactory agreement between the results of the two methods, the differences varying from 0 to 2 per cent. Again, the major item is the air hp.

30 Table 3 refers to the test shown by Fig. 11. This was made on a model compressor with a wooden casing of appreciable thickness, so that the radiation loss was practically negligible. The compressor was driven by a 75-hp., 3500-r.p.m. induction motor. Although it was a careful set-up, yet a study of the comparative results, that is, the power as found from the electrical input and that determined by the heat-balance method, will show that the difference is larger than in any of the other tabulations, the power as determined by the temperature readings being 4 per cent too

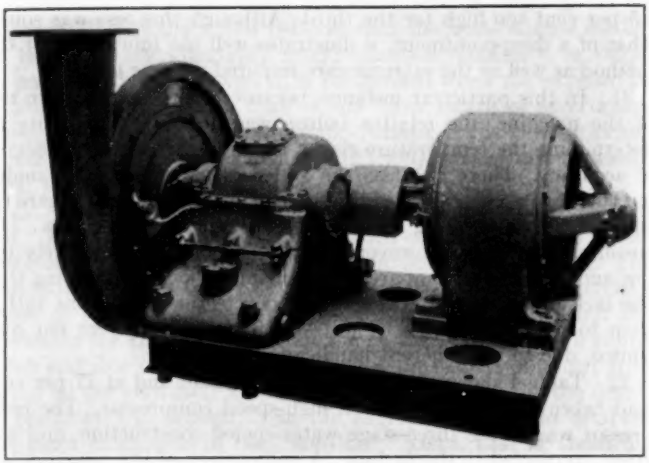

FIG. 10 CENTRIFUGAL COMPRESSOR DRIVEN BY INDUCTION MOTOR THROUGH STEP-UP GEAR

(Test results shown in Table 2.)

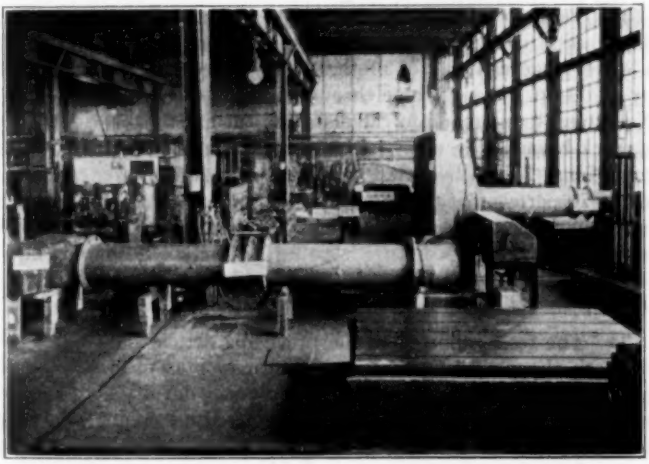

FIG. 11 TESTING CENTRIFUGAL COMPRESSOR BY HEAT-BALANCE METHOD — ASSEMBLY ON MODEL TEST COMPRESSOR WITH WOODEN CASING

low for the first point, 0.6 per cent too high for the second, and 3.5 per cent too high for the third. Although this test was somewhat of a disappointment, it illustrates well the limitations of the method as well as the extreme care required in some cases.

31 In this particular instance, because of the low pressure rise of the machine, the relative failure was due to the inability of determining the temperature rise of the air with a sufficient degree of accuracy. There was also the added uncertainty of the motor output at the comparatively light load, particularly in regard to the effect of the bearing friction and windage and pump loss. The encouraging feature, however, is that the power is alternately too low and too high as compared with the motor output, proving that the lack of agreement was due to poor testing conditions rather than to wrong principles. The tests on this set-up were not continued, due to the inherent handicaps existing.

32 Table 4 shows a point near the full load and at 77 per cent load taken on a turbine-driven high-speed compressor. The compressor was of the three-stage water-cooled construction, and was direct-connected to the turbine. A comparison of the inputs determined from the heat-balance method and that calculated from the steam readings taken on the turbine again shows agreement within 2 per cent. It will be noted that the water and air horsepower are about equal, as is usual in the case of a relatively low-pressure machine of this type.

33 Table 5 is included primarily to show the energy input distribution in the case of a high-pressure machine. The unit was a seven-stage compressor of the water-cooled construction driven by a 900-hp. steam turbine through a single reduction gear, Fig. 5. No attempt was made to calculate the turbine output directly for a check because the unit was not operated under the rated steam conditions. It will be noted that the water horsepower represents about 70 per cent of the total input, showing that in high-pressure multi-stage water-cooled machines the water hp. is the item of major importance.

Conclusion

34 The tests reported in the preceding article show agreement between the power input to the compressor as measured by the heat-balance method and that determined from drivers of known efficiency usually within 2 per cent. With the progress of the art, i.e., more skill and the adoption of instruments suitable to each type of machine tested, more reliable results are being realized. Too much stress cannot be placed on the absolute necessity of using proper testing equipment, i.e., instruments sufficiently precise for the particular set of conditions. As a general rule, the low-pressure (and hence small-temperature-rise) machines are the most troublesome, as is exemplified by the test on the wooden

model compressor reported in Table 3. On the whole, after an experience of over two years and after careful study of all the relevant matter, the writer believes that the heat-balance method of testing, being inherently sound thermodynamically, is simple to understand and to explain, is economical when the expense involved is compared with that of any other means of ascertaining the power supplied to a compressor (except in the case of motor-driven sets, with no intervening gear train), and should commend itself to more widespread use.

35 The old objection to any calorific method as giving "too low" a power input is no longer founded on facts as shown by the tests cited where both positive and negative differences occur. This criticism was overcome primarily by using a properly increased value of c_p, sufficient care in measuring temperatures and due allowance for the heat lost from the casing. With reliable equipment and due caution, satisfactory results can be obtained from about 80 per cent load and over. At the lighter loads, the degree of certainty will depend upon the engineer's ability to establish the exact magnitude of the "constant" losses, that is, those due to emissivity from the casing, to bearing friction, and imperfection of the packings.

36 It might be well perhaps, in conclusion, to call attention to the fact that the entire discussion has been limited to the method of determining the power *input* to the compressor, with no mention of the compressor *output*, which of course must be known before the efficiency of the machine is established. This has been intentional for no matter what method is used to find the power input, be it steam observations or dynamometer data on a turbine, electrical input to a motor, or a heat-balance test — the output, that is, the air quantity delivered and pressure rise obtained, can be measured independently and in accordance with some one established method. The efficiency of the compressor can then be calculated as soon as an agreement is reached as to the character of the theoretical compression specified, adiabatic, isothermal, polytropic, etc. The difficulty lies only with the uncertainty of the power input. Throughout the discussion the various troubles encountered in this particular method have been indicated and the means used in overcoming them fully described.

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37 The writer wishes to thank the compressor department of the General Electric Co. for permission to publish the results of the tests, and to express his appreciation of the encouragement in the development received from Mr. W. E. VerPlanck, the designing engineer of the department. He also acknowledges the kindness of Dr. S. A. Moss, Mr. H. L. Watson, and Mr. Wm. F. Dawson, engineers with the General Electric Co. and other co-workers for assistance, particularly in the matter of constructing the electrical

temperature-indicating device, and that of Mr. T. W. Kennedy for his help in carrying on the work in general.

APPENDIX NO. 1

THEORETICAL FORMULAS AND BASIC FACTORS

38 The formulas required for the calculations are few and simple. They are listed below:

$$\text{Air-hp.} = Jc_p(T_d - T_4)w/550 \quad [a]$$

Air Flow, Theoretical Formula:

$$w = fa\frac{P_2}{P_1V_1}\sqrt{2gJc_pT_1}\sqrt{\left(\frac{P_1}{P_2}\right)^{\frac{y-1}{y}}\left[\left(\frac{P_1}{P_2}\right)^{\frac{y-1}{y}}-1\right]} \quad [a_1]$$

Air Flow, Working Formula (due to S. A. Moss):

$$w = 3.121fD^2\sqrt{\frac{P_2(P_1-P_2)}{T_1V_s}}\sqrt{1-\left(\frac{3}{2y}-1\right)(P_1-P_2)/P_2} \quad [a_2]$$

$$\text{Water hp.} = J(t_d - t_1)W/550 \quad [b]$$

$$\text{Radiation hp.} = JS(K/3600)(t_s - t_a)/550. \quad [c]$$

$$\text{Bearing hp.} \quad [d]$$

(1) Known from previous test

(2) Determined during the performance test by a calorific method:

$$\text{hp.} = Jc(t_f - t_o)q/550 + H_r = J(t_x - t_y)Q/550 + H_r$$

(3) Estimated roughly from some formula, such as Lasche's:

$$\text{hp.} = \frac{0.0931vld}{t-32}$$

$$\text{Packing hp.} = Jc_p(t_p - t_4)w_p/550 \quad [e]$$

$$\text{Total power input} = a+b+c+d+e \quad [f]$$

where the following notation is used:

a = area of flow-measuring nozzle, sq. ft.
b = ratio of specific weight of moist air to dry air, with given relative humidity. See Fig. 3
Air-hp. = horsepower absorbed by the air
c = specific heat of lubricating oil, B.t.u. per lb.
c_p = specific heat of air at constant pressure, B.t.u. per lb.
d = diameter of bearing, inches
D = diameter of flow nozzle, inches
f = discharge coefficient of flow nozzle
g = acceleration due to gravity = 32.17 ft. per sec. per sec.
H_r = hp. radiated from bearing, estimated
J = mechanical equivalent of heat = 778 ft-lb./B.t.u.
K = coefficient of emissivity of the exposed surface, B.t.u. per deg. fahr. per sq. ft. per hr. This is given by Fig. 4, or test
l = length of bearing inches
P_a = atmospheric pressure, lb. per sq. in. abs.
P_1 = the initial, or higher absolute, or impact pressure of the flow nozzle, lb. per sq. in. absolute
P_2 = the final or lower, or static pressure, in the flow nozzle, or the pressure of the region into which the jet is discharging (usually atmospheric), lb. per sq. in. abs.

$P_1 - P_2$ = pressure drop across flow nozzle, lb. per sq. in.
q = flow of oil, lb. per sec.
Q = flow of cooling water to bearing, lb. per sec.
R = the "gas" constant in 144 $PV = RT$. For dry air, $R = 53.34$; for air of any relative humidity, R is found from relation $144 \times 14.7 V_s = 520R$
Rad. hp. = horsepower dissipated into the surroundings from the compressor casing
S = exposed surface of casing, sq. ft.
T_a = temperature of atmospheric air, deg. fahr. abs.
T_d = discharge temperature of the air, deg. fahr. abs.
T_i = inlet temperature of the air, deg. fahr. abs.
T_1 = temperature of the air in high pressure end of the flow nozzle, deg. fahr. abs.
t = temperature of the bearing, deg. fahr.
t_a = temperature of the atmosphere into which the heat from the exposed surface is dissipated, deg. fahr.
t_d = discharge temperature of the cooling water, deg. fahr.
t_f = temperature of oil leaving bearing, deg. fahr.
t_i = temperature of inlet air, deg. fahr.
t_1 = inlet temperature of the cooling water, deg. fahr.
t_o = temperature of oil entering bearing, deg. fahr.
t_p = temperature of the leakage air escaping through the clearance space between the packing and compressor shaft into atmosphere, deg. fahr.
t_s = temperature of the exposed surface, deg. fahr.
t_x = temperature of the cooling water leaving bearing, deg. fahr.
t_y = temperature of the cooling water entering bearing, deg. fahr.
v = rubbing velocity of shaft in bearing, ft. per sec.
V_s = specific volume of moist air, at 14.7 lb. per sq. in. and 60 deg. fahr. $V_s = 13.09/b$
V_1 = specific volume of air at the high pressure or initial conditions of the flow nozzle, cu. ft. per lb.:

$$V_1 = V_s \times T_1/520 \times 14.7/P_1$$

Water hp. = horsepower absorbed by the cooling water
w = weight of air discharge, lb. per sec.
W = weight of cooling water being circulated, lb. per sec.
w_p = weight of the air lost through packing leakage, lb. per sec.
$y = c_p/c_v = I/(1 - R/Jc_p)$ = the ratio of specific heat at constant pressure to that at constant volume = exponent of adiabatic expansion.

These formulas are briefly discussed below:

39 *a Air Hp.* This formula is well known. In using it, however, care must be taken to choose a proper value of c_p from Fig. 1, with due account of the effect of humidity. At usual atmospheric temperature, c_p for dry air is taken as 0.242. This is about 2 per cent higher than the value 0.2375 usually given by textbooks. The latter value is due to Regnault. The value of 0.242 is considered more reliable and is the result of the more recent experiments by W. F. G. Swann reported in Proc. Roy. Soc., 1909.[1] (According to later physicists, certain features of Regnault's apparatus had the tendency of making his deter-

[1] "The Specific Heat of Air and Carbon Dioxide at Atmospheric Pressure by the Continuous Electrical Method," W. F. G. Swann, Proc., Roy. Soc., vol. 82, Series A, 1909.

minations too low.) The slight variation of c_p of dry air with pressure as found by Holborn and Jacob [1] may be neglected in the present work (see Fig. 2) up to pressures of 100 lb. per sq. in. The influence of humidity must however be always taken into account. At 100 deg. fahr. the increase in c_p due to the presence of moisture may be as much as 4 per cent in case of high relative humidity. This is caused by the fact that moisture in the atmosphere exists in the form of superheated steam, at a pressure varying up to about 1 lb. per sq. in. absolute (at 100 deg. fahr.). Since steam is only about $\frac{5}{8}$ as heavy as air at the same pressure and temperature, and has a specific heat at constant pressure of roughly double that of air, both the density and specific heat of moist air will differ from dry air: the density is less, as shown on Fig. 3, and the specific heat is greater, as shown on Fig. 1. These figures were obtained by applying Dalton's law of partial pressures to the mixture of air and steam, using Marks and Davis' Steam Tables to get the properties of steam.

40 The second item in the computation of air hp. is the air quantity. This is usually measured by means of a flow nozzle, and can be obtained from formula a_1 or a_2. These are both based on the well known thermodynamic formula for adiabatic flow. Formula a_1 is expressed by the writer in this particular form for convenience in studying the effect of variation of specific heat, exponent and density. For exact work, the formula should be used with proper values of c_p, V_1, and y. For the usual case, however, formula a_2 given by S. A. Moss [2] is quite satisfactory. The formula will hold up to a pressure ratio P_2/P_1 as low as 0.527, and is much easier to use. The discharge coefficient f is about 0.98 — 0.99 for nozzles of streamline section having a diameter of 2 in. or over.

41 The exponent y for dry air, corresponding to $c_p = 0.242$ and the air *gas constant* 53.34, comes out abut 1.395, as compared with the value 1.40 — 1.41 usually quoted. Taking saturated air at 100 deg. fahr. as the extreme case, it can be shown by a simple calculation that the presence of moisture will reduce the value of y by about 0.01. Under ordinary circumstances, the effect on flow of the decrease in y due to moisture is practically negligible.

42 The formulas for flow as given are based on the assumption that the air is being drawn from the atmosphere. In the rare case when the intake is from a receiver, T_a, P_a and the humidity conditions refer to those of the receiver.

43 *b Water Hp.* The formula for this is self-evident.

44 *d Bearing Hp.* No general formula can be given for this. The reader is referred to standard treatises on the subject, such as Stodola's or Goudie's, etc. Whenever possible, the bearing friction should be established by specific tests. In such cases, care must be taken to use oil of the same viscosity and at the same temperature during compressor tests as obtained during the bearing tests proper. The expression given in *d* (2) will apply when a calorific test is made on the bearing loss, i.e., observations made of the heat energy carried off by the oil, by the circulating cooling water, and that lost into the surroundings by radiation and convection. The rough formula suggested is that due to O. Lasche and should show the order of magnitude of the loss.

45 *e Packing Hp.* This formula is of the same type as *a*. Whenever possible, t_p is measured. The amount of leakage depends so much upon the specific design that no formula can be given. The reader should

[1] *Zeit. Ver. Deut. Ing.*, 58, p. 1429, 1914.
[2] The Impact Tube, Trans. A.S.M.E., 1916.

consult a standard treatise like Steam Turbines by Stodola or Goudie. Sometimes, a sufficiently close approximation may be secured by replacing the annular clearance by an equivalent orifice with a discharge coefficient of about 0.50, and hence estimating the leakage flow (w_p) in the usual manner. In a properly designed machine the effect of this leakage flow is usually negligible.

APPENDIX NO 2

REPRESENTATIVE TEST RESULTS

TABLE 1 TEST RESULTS ON A 75-HP. MOTOR-DRIVEN COMPRESSOR
(Compression, 3 lb. per sq. in.)

Date of test	2/17/25	
Actual diameter of flow nozzle, in.	6.000	
Test point no.	1	2
Per cent rated load of compressor	102.8	115.0
Barometer, lb. per sq. in. abs.	14.76	14.67
Inlet temperature by resistance thermometer, deg. fahr.	71.34	71.44
Discharge temperature by resistance thermometer, deg. fahr.	115.56	115.13
Temperature rise, deg. fahr.	44.22	43.69
Air temperature preceding flow nozzle, deg. fahr.	113.4	113.3
Pressure drop across nozzle, in. water	30.24	37.95
Quantity of air, lb. per sec.	5.16	5.73
Relative humidity, per cent	45	45
Specific heat of air at constant pressure c_p	0.243	0.243
Air hp.	78	86.1
Bearing hp.	Charged to driver	
Radiation hp.	2.1	2.1
Packing hp.	Negligible	
Energy distribution, per cent of input:		
Air hp.	97.4	97.6
Radiation hp.	2.6	2.4
Input in hp., heat-balance method	80.1	88.2
Input in hp. from motor input and motor efficiency	82.2	89.5
Difference, per cent	−2.6	−1.4

TABLE 2 TEST RESULTS ON A 175-HP. GEAR-DRIVEN MOTOR COMPRESSOR
(Compression, 6 lb. per sq. in.)

Date of test	11/21/24				
Actual diam. of flow nozzle, in.	6.000				
Test point no.	1	2	3	4	5
Per cent rated load of compressor.	100	115	130	160	86
Barometer, lb. per sq. in. abs.	14.67	14.67	14.67	14.67	14.67
Inlet temperature by resistance thermometer, deg. fahr.	76.04	76.14	76.94	78.4	73.94
Discharge temperature by resistance thermometer, deg. fahr.	159.6	159.5	159.06	158.73	159.4
Temperature rise, deg. fahr.	83.56	83.36	82.12	80.33	85.46
Air temperature, at nozzle, deg. fahr.	153.53	153.53	153.53	153.53	153.53
Pressure drop across nozzle, in. water	23.06	30.03	40.25	60.0	17.01
Quantity of air, lb. per sec.	4.33	4.94	5.7	6.97	3.73
Relative humidity, per cent.	51.0	47.0	47.5	45.0	45.0
Specific heat c_p	0.244	0.244	0.244	0.244	0.244
Air hp.	125.0	142.2	162	193.2	110
Packing hp. loss	Negligible				
Radiation hp. loss	3.0	3.0	3.0	3.0	3.0
Bearing hp.	Charged to driver				
Energy distribution, per cent of input:					
Air hp.	97.7	97.9	98.2	98.5	97.3
Radiation hp.	2.3	2.1	1.8	1.5	2.7
Input in hp. from heat-balance method	128	145.2	165	196.2	113
Input in hp. from motor input.	131.2	146.7	165.2	197.3	113.7
Difference, per cent	−2.4	−1.0	−0.1	−0.5	−0.6

TABLE 3 TEST RESULTS ON A MODEL COMPRESSOR

Date of test	12/24/24	12/26/24	12/26/24
Actual diameter of flow nozzle, in.	8.003	8.003	8.003
Test point no.	1	2	3
Per cent rated load of compressor	55	56	70
Barometer, lb. per sq. in. abs.	14.555	14.965	14.74
Inlet temperature by resistance thermometer, deg. fahr.	71.32	66.51	68.00
Discharge temperature by resistance thermometer, deg. fahr.	108.59	107.62	106.14
Temperature rise, deg. fahr.	37.27	41.11	38.14
Air temperature at nozzle, deg. fahr.	106.92	102.3	100.96
Pressure drop across nozzle, in. water	4.88	4.49	8.82
Quantity of air, lb. per sec.	3.67	3.58	5.00
Relative humidity	52.0	38	33
Specific heat c_p	0.2434	0.2430	0.2427
Bearing loss		Charged to driver	
Radiation-hp. loss	0	0	0
Packing loss	0	0	0
Input in hp. from heat-balance method	47.25	50.7	65.5
Input in hp. from motor input	49.3	50.4	63.3
Difference, per cent	−4.1	+0.6	+3.5

TABLES 4 AND 5 TEST RESULTS ON TURBINE-DRIVEN COMPRESSOR

Type of unit	Table 4 Direct driven		Table 5 Driven through reduction gear
Date of test	5/21/24	5/21/24	2/26/25
Diameter of flow nozzle, in.	8.003	8.003	6.015
Test point no.	2	1	2
Per cent rated load of compressor	77	100	110
Barometer, lb. per sq. in. abs.	14.65	14.65	14.44
Inlet temperature by resistance thermometer, deg. fahr.	70.7	71.4	61.48
Discharge temperature by resistance thermometer, deg. fahr.	140.9	145.9	199.13
Temperature rise, deg. fahr.	70.2	74.5	137.65
Air temperature at nozzle inlet, deg. fahr.	137.2	142.0	193.75
Pressure drop across nozzle, in. water	12.98	22.26	38.01
Quantity of air, lb. per sec.	5.86	7.64	5.38
Relative humidity, per cent	50	50	48
Specific heat c_p	0.243	0.243	0.243
Air hp.	142.0	196.0	255.0
Radiation hp. loss	2.14	2.24	5.68
Water discharge temperature avg. of 3 thermometers, deg. fahr.	67.13	67.21	67.07
Water inlet temperature avg. of 3 thermometers, deg. fahr.	55.73	55.66	40.35
Weight of water, lb. per sec.	7.67	7.61	17.9
Water hp.	123.6	124.3	676.0
Bearing and gear loss hp.	6.3	6.3	45
Packing loss hp.	0.7	0.7	Negligible
Energy distribution, per cent of input:			
Air hp.	51.6	59.5	25.9
Water hp.	45.0	37.7	68.9
Radiation hp.	0.8	0.7	0.6
Bearing and gear loss hp.	2.3	1.9	4.6
Packing loss hp.	0.3	0.2	0
Input in hp. from heat-balance method	274.7	329.7	982
Input in hp. from turbine output	269.5	326.3	
Difference, per cent	+1.9	+1.0	

APPENDIX NO. 3

ADDITIONAL TEST EVIDENCE

From the preceding discussion it should be clear that the application of the heat-balance method to testing machines of low pressure rise involves considerable difficulties due to the influence of errors in temperature measurements and in the estimates of heat losses from the casing. In view of this, the excellent results of the latest check test reported in Table 6 furnish good proof of the reliability of the equipment used and the methods of calculation adopted. The compressor was of the general type shown by Figs. 9 and 10. It was driven by a 2-pole, wound-rotor induction motor. The losses of the motor were determined both before and after the performance tests on the compressor.

TABLE 6 TEST RESULTS ON A 145-HP. MOTOR-DRIVEN COMPRESSOR

Date of test	11/5/25				
Diameter of flow nozzle, in	10.000				
Test point no.	1	2	3	4	5
Frequency, cycles per sec	50	50	50	50	60
Per cent rated load of compressor	112.8	102.3	90.0	73.5	109.2
Barometer, lb. per sq. in. abs.	14.93	14.93	14.92	14.92	14.91
Discharge pressure, lb. per sq. in. gage	2.500	2.539	2.570	2.604	3.670
Inlet temp. by resistance thermometer, deg. fahr.	75.94	76.59	76.32	76.57	78.18
Discharge temp. by resistance thermometer, deg. fahr.	106.62	107.39	107.40	108.72	122.27
Temperature rise, deg. fahr.	30.68	30.80	31.08	32.15	44.09
Quantity of air, lb. per sec	19.4	17.7	15.6	12.8	22.4
Relative humidity, per cent	52.0	52.0	53.7	50.0	45.0
Air hp.	205.5	188.3	167.7	142.1	341.0
Bearing hp.	Charged to driver				
Radiation hp.	3.2	3.3	3.4	3.4	4.6
Packing hp.	Negligible				
Input in hp., heat-balance method	208.7	191.6	171.1	145.5	345.6
Input in hp. from motor input and motor efficiency	206.0	190.9	172.8	146.5	345.0
Difference, per cent	+1.3	+0.4	−1.0	−0.7	+0.2

APPENDIX NO. 4

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DISCUSSION

A. PETERSON.[1] This paper, so far as the writer knows, is the first publication in the English language covering the heat-balance method of testing centrifugal compressors. As the author says, this method has been used abroad. Incidentally it has been rather completely described in Ostertag's book on Kolben- und Turbo-Kompressoren, which also gives results of tests, the heat-balance method agreeing very closely with other accurate means of measuring the output from the driver. The company with which the writer is identified used this method of testing centrifugal compressors as far back as ten years ago with very satisfactory results.

In estimating the power input to the compressor the author has neglected to take into account an item which on high-pressure compressors does not amount to much, but which on low-pressure blowers may affect the results quite materially. This item is the energy in the air or gas in the form of velocity. For instance, if there is a velocity of 100 ft. per sec. in the discharge pipe of a blower for 2 lb. per sq. in. pressure, the velocity head is approximately 0.089 lb., so that an error of practically 4.5 per cent would result if the velocity energy were neglected. The method of testing blowers and compressors for efficiency as described by the author can be used only when means are available for measuring the volume delivered. Where it is not possible to obtain the volume, the efficiency for uncooled compressors can still be calculated approximately from only the temperature rise by means of the formula

$$\eta_{ad} = \frac{t_{1/2} - t_1}{t_2 - t_1}$$

where

η_{ad} = efficiency based upon adiabatic compression
t_1 = initial temperature
$t_{1/2}$ = theoretical end temperature based upon adiabatic compression
t_2 = actual end temperature.

[1] Engineer, De Laval Steam Turbine Co., Trenton, N. J. Mem. A.S.M.E.

With this method allowance also must be made for radiation, leakage, velocity, energy, etc., and it does not apply to water-cooled compressors. Suction and discharge pressures naturally have to be known when the above method is used.

These items are not given exactly in the author's test, but assuming that the figures for compression given in Tables 1 and 2, Appendix No. 2, represent fairly closely the discharge pressure for 100 per cent load, an approximate check can be made. Taking, for instance, test No. 1 in Table 1, 3 lb. discharge pressure and 14.7 lb. absolute inlet pressure gives a theoretical temperature rise based upon 71.34 deg. fahr. inlet temperature, of 28.7 deg. fahr. The actual temperature rise being 44.22 deg. fahr., the adiabatic efficiency is

$$\frac{28.7}{44.22} = 64.9 \text{ per cent.}$$

The theoretical air horsepower is obtained from formula (a) in Appendix No. 1, using $T_d - T_i = 28.7$. The theoretical air horsepower is 50.7, giving an actual air horsepower of $\frac{50.7}{0.649} = 78.1$, as compared with 78 by the author's method. The agreement between the two methods is accordingly very close.

While the method just described does not permit of estimating the power input to the blower unless the volume is known, it may, however, be of value because in the majority of blower installations, where no means are available for measuring the volume, it gives the engineer a means of estimating the approximate efficiency of the machine without much trouble and expense.

Sanford A. Moss.[1] This method is not new, but early users of the method either made inaccurate allowances for the radiation horsepower or made no allowances at all. In such early work erroneously high values of efficiency were obtained, the errors ranging from 5 to 10 per cent. The fact that a compressor has an appreciable radiation is shown by the fact that it can actually be operated at no load with an appreciable power input, while there is no heat discharged in the air because no air is discharged. Under such conditions there is nevertheless only a moderate rise in the temperature of the compressor casing, showing that there is a definite amount of radiation. The writer believes that the author is the first person to take measurements with the necessary precision and to take all factors into account in this method.

[1] Engineer, Thomson Research Laboratory, General Electric Co., West Lynn, Mass. Mem. A.S.M.E.

W. H. CARRIER.[1] In the centrifugal compressors used in refrigeration there is an evaporator which gives a definitely measured quantity of gas, known from its latent heat and the quantity of water being cooled, measured as accurately as possible. The heat extracted thus is known and the corresponding quantity of gas given off at an accurately measured pressure can be determined. Since there is a definite temperature for this gas and a sufficiently large volume, very little change of temperature is possible, due to radiation. Allowance can be made for such radiation as occurs, and the temperature of the discharged gas can always be ascertained very accurately. The quantity of heat extracted in the condenser can also be measured with great accuracy. The difference between the two quantities is the power input. If the measurements are made carefully enough, this difference in heat extracted measures the power load with great accuracy. The temperature rise of the gas and the quantity of gas as measured with the orifice method or the pitot-tube method, gives the other check. If in addition the power input is measured, there are three entirely independent methods of check. The writer has had occasion to take such measurements in centrifugal-compressor systems of refrigeration and has found the results to check accurately with the heat balance of the author. The writer thinks that the author's method of tests can be accepted with assurance for practical use if the properties of the gas compressed are known with sufficient accuracy.

THE AUTHOR. In reply to Mr. Peterson's remarks, the author would point out that with the set-up as described the error due to the supposed omission of the kinetic energy of the fluid flowing through the discharge pipe either does not exist at all or is insignificant. All of the usual temperature-indicating instruments placed in a stream of fluid actually register almost the full "dynamic" temperature, i.e., the true or static temperature plus the temperature increase caused by the energy of flow destroyed when the molecules in the immediate vicinity of the device come to rest. The set-up involves the use of similar temperature-indicating devices in both inlet and discharge, and hence the effect of velocity of flow on the temperature reading in the discharge pipe would be practically offset by a similar effect in the inlet pipe; so that calculations carried out in accordance with the scheme outlined yield correct results. As Mr. Peterson mentions, this item, the kinetic energy of the fluid, is insignificant in the case of the high-pressure machines. A serious difficulty in applying the method to low-pressure machines lies in the extreme accuracy required in measuring the temperature rise. The method is therefore not recommended for such cases.

[1] President, Carrier Engineering Corpn., Newark, N. J. Mem. A.S.M.E.

Professor Ostertag's treatment of the calorific method of testing centrifugal compressors is sufficient for the purposes of a textbook, being indicative of the general principles involved. It could not, and does not, contain sufficient details about the various factors to enable one to use the scheme with success in actual practice. Considerable work and study was involved in developing proper measuring devices, selecting proper values for the fundamental constants, designing some of the formulas, devising the method for checking the casing heat-loss coefficient under the actual radiation conditions of the test, etc. Incidentally, the book referred to implies the use of a radiation coefficient which seems too low, being about 50 per cent of what the author would recommend.

The method mentioned by Mr. Peterson of getting the efficiency of a low-pressure blower without measuring the volume is very convenient for estimating the performance of an uncooled machine. One must of course keep in mind the approximation involved.

While the subject matter of the paper was confined primarily to compressors handling air, the same general method will of course apply to a machine handling gas or any fluid, provided the necessary properties of the fluid are known, as brought out by Mr. Carrier's remarks.

No. 1992

TWO METHODS USED FOR THE DETERMINATION OF THE GASOLINE CONTENT OF ABSORPTION OILS IN ABSORPTION PLANTS

By George Stuart Clark,[1] Stanford University, Cal.
Junior Member of the Society

PART I INTRODUCTION

DURING the last decade there has been a remarkable growth in the natural-gas gasoline industry. The increase in the demand for gasoline directed attention to the recovery and utilization of those vapors which may be absorbed from natural gas or which may be recovered by compression and condensation from the gas. This condensate or natural-gas gasoline, owing to certain inherent characteristics, cannot be used efficiently for motor fuel. It is possible to blend such gasoline with products refined from crude oil which are usually too low in gravity to be marketed as motor fuel or gasoline. When the two products are properly blended an efficient motor fuel is the result.

2 The investigation described in this paper deals with the absorption process in which light hydrocarbon oil is employed as an absorbing medium. This absorption process is more largely used in California oil fields as compared with the compression method, owing to greater simplicity, lower cost, easier control, fewer operating difficulties and the greater recovery possible when natural gas is comparatively dry.

3 For testing the efficiency of the absorber two methods are commonly used in the field, one known as the activated carbon test and the other employing the "Newton" tester, neither being without faults and limitations. Some such test, applied to both the rich gas entering the absorber and the lean gas leaving it, is necessary in order to determine how efficiently the gas has been stripped of its gasoline content in the absorber. These tests do not give any information concerning the efficiency of the still whose function is to remove the gasoline content from the ab-

[1] Mechanical Engineer, Stanford University, Cal.

Abridgment of thesis submitted to Stanford University for degree of Mechanical Engineer, 1924. Awarded Student Prize at the Annual Meeting, New York, Dec. 1 to 4, 1924, of The American Society of Mechanical Engineers.

sorbent oil. Further, the fractionator must be efficient and under effective control so that a minimum of gasoline fractions will be thrown down and returned to the still for redistillation, and also so that none of the heavy fractions pass over to the condenser raising the end point of the gasoline above that permissible.

Object of the Investigation

4 The problem in hand is to compare under identical conditions the two methods commonly used for the determination of the efficiency with which the still strips the "fat" absorption oil of its gasoline content. The object of such a study is to show the practicability and the limitations of either method. The first method involves a measurement of the vapor pressures and the second method fractional distillations of the samples.

5 If the results of tests on "fat" and "lean" oils are representative a comparative record of their gasoline content may be kept. Such a record leads to effective control of still operation, and also to better control of other individual processes involved, i.e., absorption and fractionation.

Explanation of Terms Used

6 *"Fat" and "Lean" Absorption Oils.* The term "fat" oil will designate the absorption oil which has passed through the absorber and contains the natural-gas gasoline. "Lean" oil is the oil which has passed through the still and has had practically all of the lighter fractions removed. "Lighter fractions" designates those fractions used in making natural-gas gasoline, commonly known as casinghead gasoline.

7 *Gasoline Content.* The term "gasoline content" is used freely in connection with the fat and lean absorption oils. The word "gasoline" has a much broader and more general meaning today than it had some ten years ago. Today it is common to designate as gasoline any mixture or blend of oils which passes certain specifications. "Gasoline content" will mean, first, the natural-gas gasoline in the fat absorption oil; second, the light fractions in the lean oil which remain in the oil after the main distillation.

8 *"Wet" and "Dry" Natural Gas.* The gas treated by the absorption and the compression processes is usually classified as either "wet" gas (which comes from a well producing oil), or as "dry" gas (which comes principally from a well producing gas only). However, the classification is more generally based on the percentage of gasoline in the gas. Wet natural gas, or casinghead gas, contains more than one gallon of gasoline per 1000 cu. ft. of gas. Wet gas can be treated profitably by the compression method as well as by the absorption process. Dry gas, containing less than

one gallon per 1000 cu. ft., has been treated only on a commercial basis by the absorption process.

PART II FUNDAMENTAL CONSIDERATIONS[1]

9 When natural gas in the earth comes in contact with petroleum, those fractions of the petroleum having the lower boiling points will be taken up into the gas first, as their vapor pressures are much higher than those of other fractions. This vapor of the light fractions is carried along with the gas. The quantity of gasoline carried increases with an increase in the temperature of the gas. The quantity is also dependent upon the accessibility of the gas to the petroleum.

10 This gasoline which is carried along as vapor in the natural gas may be removed in several different ways. In this paper the absorption method is under consideration. To clarify the discussion which will follow a brief description of the absorption process is given.

The Absorption Process

11 A diagrammatic sketch of a modern plant is shown in Fig. 1. It is difficult to say that any one plant is truly representative of all the different absorption plants in existence. The principle, however, is the same for all.

12 The process consists essentially in bringing the natural gas, under more or less pressure, in contact with an oil heavier than gasoline flowing counter current to it and allowing the oil to absorb the gasoline vapors from the gas, after which the absorbed gasoline is removed from the oil by distillation. There is a continuous circuit of absorption oil; that is, the oil is used to absorb more natural-gas gasoline after it has had the gasoline extracted by distillation.

13 As will be seen in Fig. 1 there are many pieces of auxiliary apparatus, such as fractionators, condensers, heat exchangers, oil coolers, etc. The heat exchangers permit the conservation of much heat that might otherwise be wasted.

Natural Gas[2]

14 "In the United States most of the natural gas produced contains members of the paraffin series of hydrocarbons heavier than methane. The ordinary combustion analyses of natural gas show only the two predominating paraffin hydrocarbons, which are usually methane and ethane. The authors (Burrel, Biddison, and Oberfell) have never tested natural gas that showed methane and

[1] See Burrel, Biddison, and Oberfell, Extraction of Gasoline from Natural Gas by Absorption Methods. U. S. Bur. Mines, Bull. No. 120.

[2] The following paragraphs are quoted from pages 9 and 16, respectively, of U. S. Bur. Mines, Bull. No. 120.

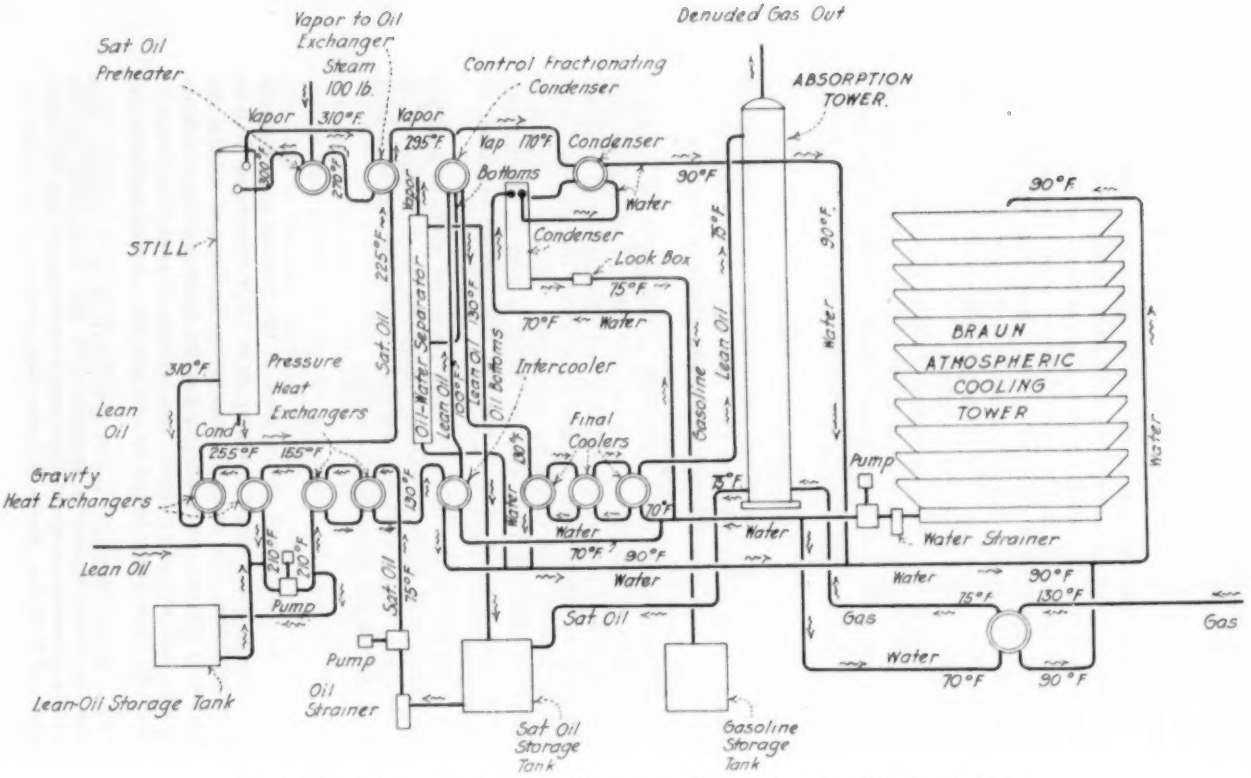

FIG. 1 FLOW DIAGRAM OF BRAUN ABSORPTION PLANT, PRESSURE DISTILLATE TYPE

ethane without finding the higher-paraffin hydrocarbons, such as propane, butane, pentane and hexane — when a test was made for them. Most of the gasoline that is obtained by the absorption method comes from the pentane."

15 "Methane, ethane, propane, and butane are gases at ordinary temperatures. Pentane and hexane are liquids at ordinary temperatures, and are those vapors carried by the gases that constitute most of the gasoline extracted from natural gas. Some butane and, in compression plants, some propane are also ex-

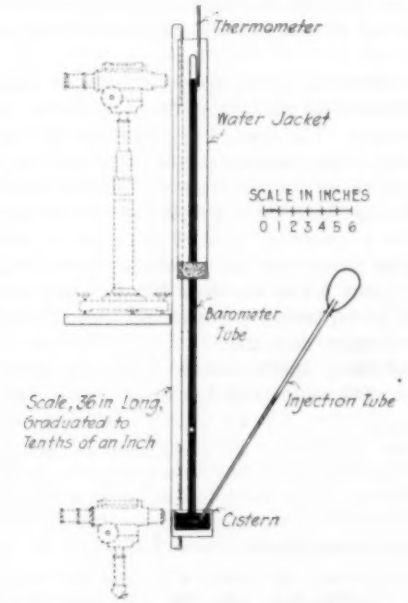

FIG. 2 VAPOR-PRESSURE APPARATUS NO. 1 — THE CISTERN BAROMETER

tracted. These are chiefly responsible for the "wildness" or high vapor pressure of some gasoline extracted by compression and condensation."

ABSORPTION OIL

16 There is a great variation in the properties of absorption oils. This is due to the degree to which they are refined and to the source from which the stock came. The oil generally used is that product which is distilled from crude oil after all of the water white products, including heavy kerosene, are off. This oil is known as "mineral seal" oil, or as "straw" oil.

17 The properties to be considered in selecting oil are,

a Vapor pressure or tension

b Initial boiling point and

c Viscosity at different temperatures.

18 Vapor tension of the absorption oil should be low, in order that the maximum amount of gasoline may be absorbed in a given amount of oil. When the vapor pressure of the oil due to its gasoline content approaches the vapor pressure of the gasoline in the natural gas, the capacity of the oil to absorb decreases rapidly.

19 The initial boiling point of the absorbing oil should be considerably higher than the end point of the gasoline so that as little of the heavier oil as possible will be carried over into the fractionator.

20 The viscosity of an absorption oil deserves some consideration. The oil must be manipulated through the towers, baffles, piping, and pumps. The ease with which the oil may be worked is increased when the viscosity is low, especially at low temperatures which are to be desired from the standpoint of absorption.

21 Considering these three properties the absorption oil to be used should be a "narrow" cut of oil; that is, one with initial and final boiling points not too widely separated. An oil with an initial boiling point not lower than 450 deg. fahr. should be used. An oil with an initial boiling point of not lower than 500 deg. fahr. would be much more desirable.

22 The following specifications define an oil which is well suited for this work and which has been used for this purpose.[1]

Flash point	247 deg. fahr.
Fire test	262 deg. fahr.
Specific gravity	37 deg. B.
Over point	450 deg. fahr.
End point	695 deg. fahr.

The distillation temperatures are as follows:

Per cent	10	20	30	40	50	60	70	80	90	100
Deg. fahr.	528	540	556	565	572	572	595	608	633	695

23 The absorption oils used in the tests described in this paper are compared to the above oil in Table 13.

PART III APPARATUS

Vapor-Pressure Apparatus No. 1[2]

24 *Description.* The vapor-pressure apparatus No. 1 used for determining the gasoline content of absorption oils is shown in Fig. 2. This apparatus differs only slightly from the apparatus

[1] W. P. Dykema, Recent Developments in the Absorption Process for Recovering Gasoline from Natural Gas. U. S. Bur. Mines, Bull. No. 176, page 33.

[2] Used in practice.

used in practice. The differences are in the shape and size of the water jackets, which are of secondary importance so far as the accuracy of results is concerned.

25 As shown, the apparatus was a water-jacketed cistern barometer. The cistern barometer consisted of a straight hard glass tube, $\frac{7}{16}$ in. inside diameter, about 33 in. long, sealed over at one end, filled with mercury, and dipping into a cistern containing the same metal. The water jacket used was a glass tube connected to the barometer tube by means of a large rubber stopper. The jacket was 16 in. long and placed on the tube so that at the highest vapor pressures all of the space above the mercury and at least four inches of the mercury would be jacketed. The water in the jacket, when properly stirred, insured good temperature control while the readings were being taken.

26 A 36-in. scale, graduated to 0.1 in., was hung vertically beside the water jacket and tube in order to make the necessary readings. It was so hung that readings could be made of the height of mercury in the cistern as well as the height in the tube. A cathetometer microscope was used, permitting readings to be made to 0.01 in.

27 *Procedure and Discussion.* At room temperature the vapor pressure of a mixture of gasoline and absorption oil is less than that of the atmosphere. In order to measure it with this type of apparatus it is necessary to inject a small quantity of the liquid into the bottom of the tube of the barometer set up as shown in Fig. 2. This was done in these tests by means of a bent glass tube and a rubber bulb, working on the principle of a medicine dropper. This part of the apparatus is shown in the proper position for injection. The long injection tube was calibrated so that an approximately equal volume of absorption oil could be injected each time. This precaution was necessary to insure that a sufficient quantity of the liquid was present to give a dependable value of the vapor pressure. Furthermore, it was desirable to have equal volumes injected for the sake of uniformity in data and corrections.

28 After the absorption oil is injected it rises in the tube and starts to vaporize even before it reaches the top of the mercury column. The pressure of this vapor will cause the barometer to fall. The change in height of the column will be due to two pressures:

a The pressure exerted by the vapor from the mixture of oil and gasoline, pentane and hexane, the vapors in natural gas that constitute most of the gasoline extracted, are liquids at ordinary temperature and atmospheric pressure and are the constituents in the mixture that should be measured by the apparatus. However, some butane, and in casinghead plants, some propane, are also extracted. If they are included in the sample, even in small

amounts, they will affect the results greatly because of their very high vapor pressures.

b The pressure exerted by any air that may have been absorbed in the mixture before its injection, or the pressure exerted by any air that may not have been removed from the mercury column and which may be collected or carried along by the oil, from the walls of the tube, during its ascent in the tube.

29 *Recording.* Before the sample of oil was injected the temperature of the jacket water was 56 or 57 deg. fahr. After the oil had been injected and had risen to the top of the column, it was allowed to "settle" until there were no bubbles visible in the oil. The jacket water was stirred intermittently and when its temperature became 60 deg. fahr. the reading was made.

30 In order to insure accurate readings to 0.01 in. the cathetometer microscope was used in reading the scale. A reading was not taken unless the leveling bubble indicated that the microscope was swinging in a horizontal plane. This made it possible to make a focus on the scale without error after the microscope had been focused on the mercury meniscus. Two readings were made from the same datum for each determination; one to the top of the meniscus in the tube and the other to the top of the meniscus in the cistern. One set of these readings was made before and another after injection; they are called "initial" and "final" for recording purposes. The difference between the two initial readings was checked in each case with the reading of a standard barometer. It is understood that in practice this is not done. This care was taken in setting up so that the highest degree of accuracy would be obtained. The height of the oil on top of the mercury column and the volume of the vacuum were noted and recorded.

31 *Preparation of the Tubes.* In considering the preparation of the tubes the main object of these tests was not forgotten; that is, reliable readings must be made in order to obtain a satisfactory comparison of methods. Therefore, in filling the barometer tube some of the procedure followed in practice was altered for the purpose of these tests.

32 Two conditions which are essential to the success of any barometer are: first, cleanness and dryness of the mercury and tube; second, a good vacuum above the barometer column. With regard to the first, cleanness is imperative since indications of the barometer are dependent upon a reasonable constancy of the capillary depression of the mercury column due to the action of the surface tension at the mercury meniscus in the tube. Mercury admits of preparation in a highly purified state without great difficulty. In Appendix 3, Pars. 110 to 118, three methods of purifying mercury are described. The one used was the dilute nitric acid method (Method No. 1). In some practice the mercury is assumed to be cleaned after it has been brought to a red

heat and allowed to cool. This method is not approved by chemists since it is impossible to get a pure mercury, free from oxides, by heating alone.

33 The next important step in the preparation of the tubes was to take great care in cleaning and filling the tube. The method

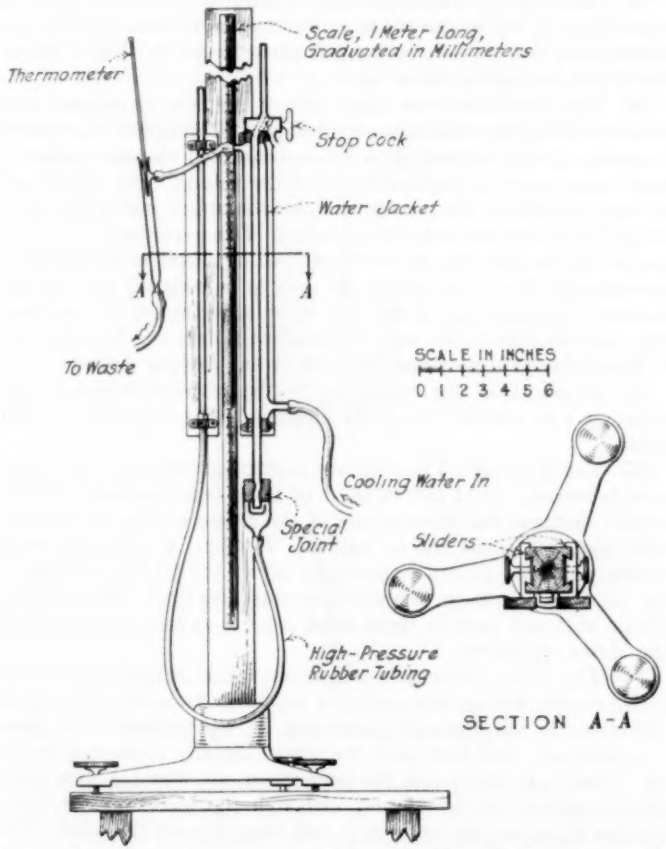

FIG. 3 VAPOR-PRESSURE APPARATUS NO. 2 — THE SIPHON BAROMETER

used in these tests, and the one most generally adopted, is that of boiling the mercury in the tube. After the tube was well cleaned and dried, clean mercury was introduced to a depth of about 10 in. and then heated. It is impossible to pour the mercury into the tube without having air, with its moisture content, collect on the sides of the tube between the walls and the mercury. By careful and persistent boiling the air bubbles are expelled. The tubes

used in these tests were filled in three stages. This method of filling by boiling gave very satisfactory results. The barometers filled in this manner checked the standard barometer within 0.04 in.

Vapor-Pressure Apparatus No. 2[1]

34 *Description.* Vapor-pressure apparatus No. 2 was used in these tests to aid in the study of the vapor-pressure method for determining the gasoline content of absorption oils. Fig. 3 shows the apparatus used in these tests.

35 The apparatus was made up of a siphon barometer and a water system for temperature regulation. The siphon barometer consisted of two vertical glass tubes, connected to each other at their lower ends by high-pressure rubber tubing. The upper end of one was open to the atmosphere and the upper end of the other had a 3-way vacuum stopcock attached. This permitted the upper end of the vacuum leg to be either open or closed to atmospheric pressure and the tubing above the cock to be emptied and cleaned easily by the third way of the cock without destroying the vacuum. The vertical glass tubes were attached to sliders which moved up or down the main post and clamped at any desired position.

36 In adapting this barometer to the tests described in this paper, it was necessary to make changes in the stopcock and the joint.

37 *The Stopcock.* The ordinary vacuum grease, used as a stopcock lubricant, could not be used in these tests because gasoline readily destroys the effectiveness of the grease so that a perfectly tight stopcock could not be realized. Further, a very inert substance which would not appreciably affect the vapor pressure of the oil was necessary to insure accurate readings. Syrup-phosphoric acid was used in these tests, since it satisfactorily fulfilled the above conditions.

38 *The Joint.* The connection between the vacuum leg and the high-pressure tubing was designed especially to adapt the siphon barometer to the needs of these tests. It was impossible to make a satisfactory joint here with the tubing directly connected to the leg. Even with shellac on the connection, air would collect when the barometer was filled; and, although the joint was air tight, bubbles of air would continue to rise from around the joint. This introduced an element of uncertainty since it was impossible to tell when no more air would rise.

39 Fig. 4 shows the joint used. This joint was so designed that any air coming from around the joint or from the walls of the rubber tubing would be trapped at the joint. The connection between the high-pressure hose *H* and the glass cup *A* was shellacked and made airtight. The rubber stopper *B* was shellacked

[1] A siphon barometer adapted to the purpose of these tests.

to the vacuum leg D of the barometer. The connection between the cup A and the stopper B was made very nearly air tight by the use of vaseline. Such a connection was advantageous because of the great number of times that the connection had to be taken apart for cleaning. The small amount of air that did pass between the walls of the cup and the stopper was trapped and did not have to be considered until the pressure of the confined air was sufficient to force the mercury level in the connection down below the end of the glass tube D. Some leaking at this joint was noticed at every reading, but even for the longest reading the pressure in the confined space increased so slowly that there was sufficient time to make accurate readings and then to check them.

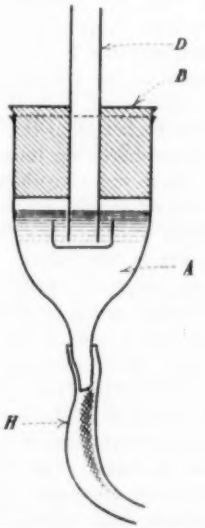

FIG. 4 JOINT OF SIPHON BAROMETER

40 For the temperature control a water jacket was put on the upper 12 in. of the vacuum leg. Tap water which had been cooled in a copper coil immersed in an ice bath was circulated through the jacket. The velocity of the water through the coil was regulated so that a constant temperature of 60 deg. fahr. was maintained. The temperature reading was the temperature of the water in the tube leaving the jacket.

41 *Procedure.* After the apparatus was filled with the proper amount of mercury, the atmospheric leg was raised, while the stopcock on the other leg was open, until the mercury level stood above the stopcock. The cock was then closed and the atmospheric leg lowered 780± mm. so that a vacuum space of considerable

size was formed in the vacuum leg. Theoretically the difference in the height of mercury in the two legs should be equal to barometric pressure. An initial reading was made for each test but this reading was always approximately 10 millimeters of mercury less than atmospheric pressure. But few of the readings taken were a millimeter more or less than the average, 757 mm.

42 The readings of mercury height on the two legs were made with reference to the datum of the scale, which was located between the two columns on the main post of the apparatus. The readings were made by holding the edge of a thin cardboard strip horizontally against the column and the scale.

43 While the water in the jacket was maintained at a constant temperature of 60 deg. fahr. the vacuum was released by leveling the mercury in the two legs. Approximately one cubic centimeter of the mixture of oil and gasoline to be tested was allowed to run in on top of the mercury in the vacuum leg. The two legs were then operated on the sliders so that the top of the oil in the vacuum leg stood a little higher than the stopcock. The cock was then closed and the atmospheric leg lowered to the proper level. With such a treatment there was considerable frothing or bubbling in the oil. After this had subsided a reading was taken. This procedure gave results which are the basis for Tables 4 and 5, upon which Curves 4 and 5, Fig. 8, are based.

44 However, in order to prove certain points which are to be discussed later an additional series of tests was made. A slightly larger sample of oil was subjected to the vacuum as described in the preceding paragraph. After frothing and bubbling had subsided the vacuum was released. Thus the saturated air and the lighter fractions of the gasoline in the sample which could not be held in the oil under the vacuum and 60 deg. fahr. were separated and forced through the stopcock. The stopcock was closed again and the oil subjected to a new vacuum. Each time the new vacuum was formed a new reading was taken. This procedure was repeated until there was no more bubbling and until three readings were obtained which checked within one millimeter. By this method of procedure the vapor pressures in Table 3 were obtained. The result in Table 3 show that with this procedure reliable and comparative vapor pressures may be obtained.

Fractional-Distillation Apparatus

45 *Description.* The still used in the tests, with several modifications, was a duplicate of some extensively used in practice in certain southern California oil fields. The body of the still was very similar to that of the still developed by the Bureau of Mines, and described in its Bulletin No. 207, page 61. The apparatus was designed to handle a charge of 1000 cu. cm. of oil. Since these

tests were started some of the companies have found it desirable to increase the capacity of the stills thus used to 2000 cu. cm.

46 *The Body of the Still.* A four-pint (1892 cu. cm.) Kjeldahl flask was used for this element of the apparatus. In practice a 2000-cu. cm. flask is used for a 1000 cu. cm. still.

47 Fig. 5 shows the general form, dimensions, and type of construction of the still and condenser. A special threaded flange was brazed on the neck of the flask. This flange is a part of the metal-

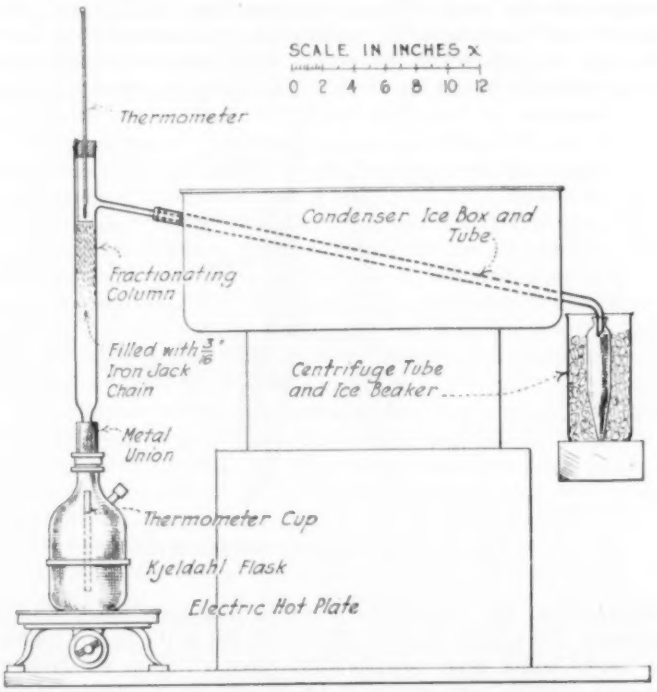

FIG. 5 FRACTIONAL-DISTILLATION APPARATUS

glass union between the still head and the flask. It was necessary to braze in the copper tube which is shown and which served as a thermometer cup. The short piece of $\frac{1}{2}$-in. copper pipe was also brazed in. This pipe opening was used for either filling or emptying the flask when it was not expedient to disturb the metal union.

48 *The Union.* The metal-glass union is shown in detail in Fig. 6. A stuffing box is resorted to for such a union since it is necessary to have the connection made tight with a material which will withstand temperatures around 600 deg. fahr. A gland

was not used in this union. The cap screwed down on the threads of the inner piece *c*, compressed the asbestos packing and made the joint tight. The inner piece *c* is attached to the neck of the flask by means of another cap *b* which screws on the threads of the neck flange. A small gasket makes a tight connection between the neck and the inner piece *c*.

49 *The Still Head.* The still head used in these tests was made of pyrex glass. The main tubing was drawn down at the lower end and welded to a piece of glass of smaller diameter which fitted snugly into the union. The main column was 18 in. long. The glass "snoot" leading to the condenser left the column 3½ in. from the top. An inch below this point, $\frac{3}{16}$-in. iron jack chains were hung vertically from a spiral wire. Enough of the chain was thus

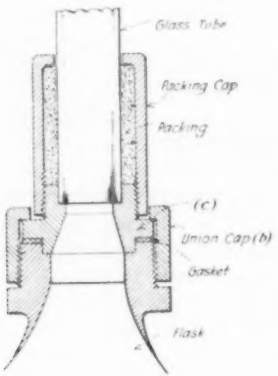

FIG. 6 METAL-GLASS UNION FOR STILL

suspended so that the column would be compactly filled. Twelve inches of the column has this filling material in it. Such a column is known either as a "fractionating column" or as a "dephlegmating column" and serves to fractionate the ascending vapors by condensing and returning the heavier fractions of oil to the still while any of the lighter vapors being carried down are reëvaporated by the ascending warmer vapors. The filling material was supported from the cork stopper at the top of the column, and through this stopper a thermometer was inserted. The bulb of this thermometer was placed ½ in. below the point of connection of the glass snoot and column. The bulb was kept covered with cotton which remained saturated at all times during a distillation. The temperature indicated by the thermometer was that of the saturated vapors with no chance of errors due to superheating. In such distillations it is not probable

that any appreciable error will arise from superheating even without such precaution.

50 *The Condenser and Receiver.* The condenser tube was a piece of ½ x 30-in. brass tubing. It was bent on the end as shown in Fig. 5. The tube passed through a galvanized iron box, 24 in. long. During a run this box was kept packed with ice. The receiver was a centrifuge tube which had an enlarged neck so that it would fit snugly over the end of the condenser tube. The receiver was surrounded by an ice bath. This was done so that as little as possible of the recovery would be lost. An asbestos ring was used to make a tight joint between the condenser tube and the receiver. Besides allowing very little of the lighter vapors to escape this ring absorbed any water which had condensed on the side of the box and had run down to the end of the condenser tube.

51 *Procedure — Distillation Apparatus.* The stills used in practice differ from the one described above in one other respect. For convenience in operating such a still the copper flask should have an outlet pipe at the bottom of the flask. This outlet pipe would pass through a hole in the hot plate and, thence, to waste. This would make it possible to pour a 1000-cu. cm. sample of oil in the pipe opening and, after the distillation is complete, to let the remaining sample discharge through the outlet pipe without disturbing the metal union. This operation permits the handling of the apparatus sooner after a distillation.

52 In these tests such an outlet was not used. The metal union was disconnected after each distillation when the oil in the flask had cooled sufficiently. The flask was then used as a receptacle to mix the next sample of oil. For a 10 per cent mixture, 100 cu. cm. of gasoline was mixed with 900 cu. cm. of absorption oil. The oil and the gasoline were poured into the flask separately and then the flask was tightly corked and shaken by hand quite vigorously for about 30 seconds. After this procedure the mixture was allowed to stand for approximately five minutes with the cork tightly in the flask. This mixture of oil and gasoline is discussed and compared to the actual samples of fat and lean oil under Discussion of Results, Part IV.

53 After sufficient time had elapsed and the oil had absorbed the gasoline the apparatus was set up as follows: The flask was placed on the electric hot plate, the still head was connected to the flask by means of the metal-glass union, and the glass snoot leading from the column was made snug in the brass condenser tube by means of a cork stopper. The glass snoot projected into the tube at least two inches. Then the centrifuge tube with an enlarged neck was placed under the lower end of the condenser tube to receive the recovery. While the oil in the flask was being heated by the hot plate, and before any of the very light vapor had

started to distill over, the condenser tube was surrounded with cracked ice and the centrifuge tube was placed in a glass jar filled with finely cracked ice. A thermometer was placed in the cup of the flask and temperature of the oil in the flask was read from it. The thermometer at the top of the fractionating column remained in the same position for all tests. It was not necessary to remove or shift it in the column.

54 After the ice was placed and every part of the apparatus connected properly, the excessive radiation losses were prevented by placing asbestos sheeting around the flask and the still head. Even with this precaution the distillation was rather slow. With the small, 1500-watt hot plate it took approximately 30 minutes for the first vapors (that would condense under the conditions) to distill over. For the distillation up to 450 deg. fahr., on the upper thermometer, 30 minutes more were necessary, but this time varied according to the percentage of gasoline in the mixture. The total time taken for setting up, distilling and collecting the condensate was between 1½ and 2 hours. This time does not include the time spent in distilling the recovery in the special apparatus.

55 The thermometer at the top of the fractionating column was carefully watched. In making such a distillation the object in view was to treat each mixture of oil and gasoline the same. That is, all of the lighter fractions which would come off below 450 deg. fahr. by use of such an apparatus were desired in the recovery. At this point in the procedure there were a number of variations in time of reading the thermometer at the top of the column. For the first run (Table 6) the hot plate was shut off when the top thermometer reached 450 deg. fahr. except in special cases noted. In the second run (absorption oil No. 2, Table 9) the heat supply was cut off at a temperature near 442 deg. fahr. and the maximum temperature at the top of the column was recorded.

56 It is at this point that this procedure in distillations is a modification from actual practice. It might be noted that in practice the specific gravity of the recovery from the sample of oil is compared with the specific gravity of the gasoline which is being produced by the plant. When the specific gravities are equal the volume of recovery is noted and compared with the volume of the test sample.

57 After the electric hot plate had been shut off the apparatus was allowed to stand two or three minutes until the condenser tube was drained. The column was then disconnected from the condenser tube and a cotton swab was rammed down the tube and removed by an attached string. This cotton collected the remaining recovery and insured a clean tube for the next distillation. The difference in weight of the cotton, before and after, was recorded by weighing to one milligram. This operation merely

showed that such a loss is of no appreciable importance and that it is a fairly constant loss.

58 The recovery from this distillation was studied in three ways:

a It was measured in a graduate calibrated in cubic centimeters

b The specific gravity was determined by means of the Chainometer

c A distillation, determining the percentage of recovery over at various temperatures, was made for each recovery. These distillations were made by means of the extra distillation apparatus described in Appendix 2, where the procedure for operating this apparatus also is given.

PART IV DISCUSSION OF RESULTS

59 In making such tests for the gasoline content of fat and lean absorption oils it is not necessary to determine the exact quantity of light fractions in the oil; nor is it necessary to know exactly what hydrocarbons are in such quantity. The object of such tests is to obtain comparative data on the operation of the still, i.e., data which can be compared to some standard of good operation. Both the vapor pressure and the distillation types of apparatus now in use are not accurate from the point of view of the chemist and physicist. The questions of absolute vapor pressures and accurate fractional distillations is of secondary importance in discussing the results. The principal question is: Which of the two methods under consideration will give consistent and comparative results for the purpose of plant control and, furthermore, under what conditions will they give those results?

60 For purposes of comparison it was necessary to make a mixture of gasoline and absorption oil for each test. By using these different mixtures in the apparatus already described the changes in vapor pressures and distillation recoveries were noted and recorded. These data form the basis of the curves to be discussed. Such a mixing under proper conditions should give a fairly representative fat oil; but at lower percentages of gasoline the mixture does not truly represent a lean oil. This is true because even under the worst conditions of operation a still will not allow the very lightest fractions of gasoline to be carried on through with the lean oil. On further consideration it may be noted that if apparatus is used to test mixtures representing lean oil it will be operating under extreme conditions. If consistent results are obtained under these conditions, the value of the tests is increased.

Cistern-Barometer Results

61 Curve sheet No. 1, Fig. 7, is a graphical presentation of the results appearing in Tables 1, 2, and 3. The first two tables are the data obtained from experiments with the cistern barometer

(vapor-pressure apparatus No. 1). Table 3 is a tabulation of data obtained by use of the siphon barometer (vapor-pressure apparatus No. 2) under best test conditions. The curves are plotted as follows: Ordinates as millimeters of vapor pressure, and abscissas as percentages of gasoline in the mixture. As may readily be seen the results obtained by means of the cistern barometer are not consistent.

62 The tests made with the cistern barometer were on mixtures which undoubtedly contained considerable quantities of air. It was hoped that the degree of saturation would be the same if all of the mixtures were treated in a similar manner. After a mixture had been stirred mechanically in a large tube it was tightly corked

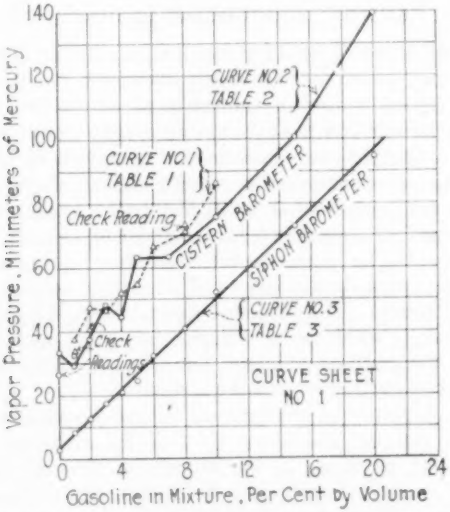

FIG. 7 CISTERN-BAROMETER RESULTS COMPARED TO BEST SIPHON-BAROMETER RESULTS

(Graphical presentation of results given in Tables 1, 2, and 3. Mixtures were of absorption oil No. 1 and 71 deg. B. gasoline.)

and allowed to stand for at least ten minutes. This procedure introduced an element of uncertainty. One of the objects of this paper is to determine whether or not this procedure, which is used in practice, is permissible; that is, whether there is a fairly constant error due to the air mixed with the oil. It is very true that in mixing a sample as described above there is chance for a larger quantity of air to become mixed with the sample than is to be found in a sample taken from a working plant. Since the oil circulating in a plant is under pressure and occasionally in contact with the atmosphere the oil is certain to contain traces of air. The amount of air contained in the oil at the plant is probably

more nearly constant but, nevertheless, any element of uncertainty such as this is not warranted unless it can be proved that the quantity of absorbed air is constant.

63 One of the advantages of the cistern barometer is the simplicity of operation. With a given sample of oil and a barometer tube which has been properly boiled out, the cistern barometer should give consistent results. There is not a step in the procedure which allows one to ascertain the composition of the sample; i.e., there is no step to tell whether any of the lighter hydrocarbons

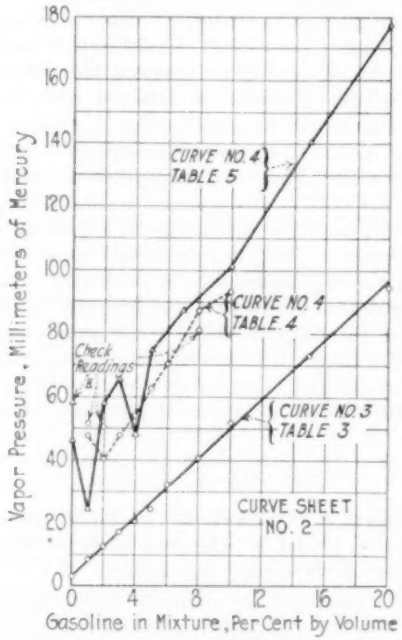

FIG. 8 SIPHON-BAROMETER CURVES

(Graphical presentation of results given in Tables 3, 4, and 5. Mixtures were of absorption oil No. 1 and 71 deg. B. gasoline.)

such as butane has been absorbed and retained by the oil. Butane when absorbed by the oil is ordinarily lost when the gasoline is allowed to weather as in some plant practice.

COMPARISON OF CISTERN-BAROMETER RESULT TO BEST SIPHON-BAROMETER RESULTS

64 It is possible to put each sample of oil in a vacuum at a certain temperature before injecting it into the cistern barometer. This vacuum could have been released and applied repeatedly until

upon shaking no more bubbles arose from the oil under vacuum. By this operation the light hydrocarbons and the air in the sample could be driven off. If the temperature is regulated properly the hydrocarbons such as pentane and hexane will remain in the sample.

65 The siphon-barometer curve (Curve No. 3, Fig. 7) was obtained by a treatment similar to the one described in the preceding paragraph and at a temperature of 60 deg. fahr. The cistern barometer was not tested under such conditions. The cistern barometer with a sample of oil treated similarly at 60 deg. fahr. would probably have given readings which would approximate the readings of the siphon barometer. This statement is partly verified by the rather rough agreement between the cistern and siphon barometer readings tabulated in Tables 1 and 2 and 4 and 5, respectively, and shown plotted in Figs. 7 and 8. These curves check roughly to about 10 mm. The siphon-barometer readings are constantly higher than the cistern-barometer readings. This is to be expected since in the cistern barometer there is a chance that some of the absorbed air in the oil will cling to the sides of the barometer tube and will not reach the top of the column and, therefore, the vapor pressure will be decreased.

66 The conclusion drawn from the first results was that neither apparatus was satisfactory unless some precaution were taken to eliminate the errors due to absorbed air and very light hydrocarbons in the sample. This led to the adoption of a new procedure with the siphon barometer and to the results tabulated in Table 3.

Siphon-Barometer Results

67 Curve sheet No. 2, Fig. 8, is a graphical presentation of the data in Tables 3, 4, and 5. The data in all of these tables were obtained by use of the siphon barometer.

68 Curves 4 and 5 of this sheet show the inconsistency of the results when the vapor pressures of samples of oil are taken without regard to the quantity of air contained in the mixture. These curves serve only to indicate that such a procedure for determining vapor pressures is too inaccurate. It is difficult to obtain a check reading even with a sample of the same mixture of oil.

69 Curve No. 3 has been mentioned and partly discussed in connection with the cistern-barometer results and siphon-barometer procedure. In obtaining these results no attempt was made to determine the absolute pressure of the mixture of oil, gasoline and absorbed air. The first vacuum set-up was broken after all frothing and bubbling had subsided and the oil was subjected to a new vacuum. By releasing the vacuum and operating the sliders it was possible to force out the air and those lighter vapors which separated under the first vacuum at 60 deg. fahr. By repeating this

operation all such air and vapors were expelled and a constant reading was obtained. These constant readings are the basis for Curve No. 3. The siphon barometer as adapted for these tests is a very convenient piece of apparatus for obtaining such results.[1]

70 In expelling the lighter vapors of the mixture an absolute vapor pressure of the sample is not obtained. It must be remembered that for the problem in hand a comparative pressure is all that is necessary to keep a good record of operation, once a standard for efficient operation has been established.

71 There is one other point in favor of such operation and readings. The hydrocarbons pentane and hexane are the chief constituents of natural-gas gasoline, and are liquids at ordinary temperatures and atmospheric pressure. In the absorption process some butane and other light hydrocarbons are usually absorbed and carried by the absorption oil to the still. A slight variation in the amount of butane in the oil will make a large variation in the vapor pressure of that oil. Butane and the other light hydrocarbons which are not liquids at ordinary temperatures will tend to be expelled by the operation just described. The vapor pressure without these lighter vapors is a more representative vapor pressure. It represents the true amount of pentane and hexane in the absorption oil. The presence of these constituents is more important than the presence of butane or propane since the latter are largely lost after the casinghead gasoline has weathered for some time.

72 The more important advantages of the siphon barometer are: (*a*) The easy manner in which a vacuum may be set up, and (*b*) the convenience with which the sample of oil may be placed in the barometer and treated.

73 Without the use of some such apparatus it would have been difficult to interpret the results obtained by the cistern barometer, and therefore difficult to make a useful comparison with the distillation apparatus. The mixtures of oil and gasoline were all treated the same in this apparatus and consistent results were obtained. Some of the secondary advantages which the siphon barometer has are: (*a*) less tubing to clean; and (*b*) the mercury column is not contaminated. The top portion of the column which comes in contact with the oil may be run out through the stopcock and separated from the clean mercury.

74 While this type of barometer has an advantage over the cistern barometer in ease of operation and preparation, there are several conditions which cannot be eliminated and which keep this type of apparatus from being a precision instrument. Without boiling out the mercury in the tubes it is practically impossible to obtain a reading of true barometric height. It is difficult to

[1] See Pars. 41 to 44 for full description of procedure.

fill a siphon barometer and boil the mercury in the tubes. Furthermore, the connection between the glass tubing and the stopcock may be rough and inaccessible to thorough cleaning; any portion of the mixture used in previous tests and left in the vacuum may affect the readings slightly. The lubricant used on the stopcock will cause a small but constant error.

Fractional-Distillation-Apparatus Results

75 The data obtained by means of the fractional-distillation apparatus may be found in Tables 6 and 9. Table 6 gives the data

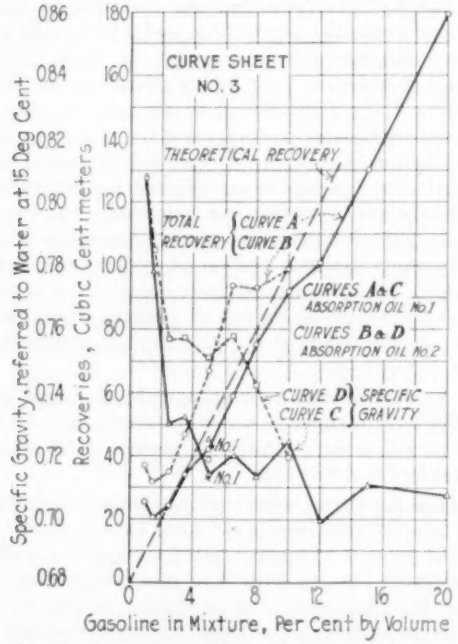

Fig. 9 Fractional-Distillation Apparatus

(Graphical presentation of results given in Table 6. Data on recoveries from mixtures of absorption oils and 71 deg. B. gasoline.)

on mixtures of absorption oil No. 1 and 71 deg. B. gasoline. Table 9 gives data on mixtures of absorption oil No. 2 and the same gasoline. The results, using these two series of mixtures, have been plotted on Curve sheet No. 3, Fig. 9.

76 Curves *A* and *C* (absorption oil No. 1) and curves *B* and *D* (absorption oil No. 2) are not the results of exactly the same procedure. The difference was in the time at which the heat supply was cut off. Curves *A* and *C* show results when the heat supply was shut off at the instant the thermometer at the top of the

fractionating column reached 450 deg. fahr. or, in the special cases, the other recorded temperature. Curves *B* and *D* are the results of recording as follows: The heater was shut off at a temperature about eight degrees below 450 deg. fahr. and the maximum temperature indicated by the thermometer was recorded. In the case of the 6½ per cent mixture the heater was not shut off until a considerably higher temperature was reached. This point is taken into account and corrected by additional treatment of results.

77 The procedure for curves *A* and *C* should theoretically give a larger recovery than the procedure for curves *B* and *D*, provided all other conditions were equal. But the results show the reverse. Both the recoveries and the specific gravities are considerably higher in the case of *B* and *D*. This is attributed to the difference between absorption oils No. 1 and No. 2.

78 For each set of curves the conditions were as nearly the same as was possible. The similarity of the two curves should be considered. At higher percentages of gasoline in the mixtures the specific gravity and volume of the recoveries approach the specific gravity and volume of the gasoline used in making the mixtures. At the low percentages, and the most important mixtures, the recovery from such an apparatus does not equal the quantity of gasoline in the mixture either in volume or specific gravity. The recovery is greater than is to be expected and a corresponding increase in specific gravity is noted. This indicates that the efficiency of the fractionating column is poor, especially at the lower percentages. In other words, the lighter fractions of oil carry over a considerable amount of the heavier fractions before the boiling point of the heavier fractions is reached. Other reasons for the apparently large mechanical carry-over are cracking, when "green" absorption oil was used, and temperature lag in the thermometer at the top of the fractionating column. At the lower percentage of gasoline in the mixture the actual recoveries are larger than the theoretical recoveries. The percentage of actual over the theoretical increases as the gasoline content of the oil decreases. The total or actual recovery from the 1 per cent mixture is greater than the actual recovery from the 1½ per cent mixture in both series of tests. This is taken as proof that the volume of the recovery from small percentages cannot be relied upon as an indication of the gasoline content of an absorption oil when such apparatus is used and conditions are reasonably constant.

79 The specific gravity, on the other hand, is a fair indication at these smaller percentages of gasoline content. On referring to curves *C* and *D* a large increase in specific gravity of those recoveries from the 2½ per cent to 1 per cent mixtures is evident. The specific gravity of the recovery approaches the specific gravity of the oil in the sample. It would be possible to record without considerable error the gasoline content of the very lean oils from the

recovery specific gravity, provided some standard of comparison was established. But when fat oil was tested the recovery specific gravity would not give an accurate indication.

80 In order that the conditions of the test might be studied fully, distillations of the recoveries were made by use of additional apparatus. By making a distillation of each recovery and noting the temperatures for certain percentages over, it was possible to check completely the data obtained with the fractional distillation apparatus. For the results of these distillations see Tables 7 and 10,

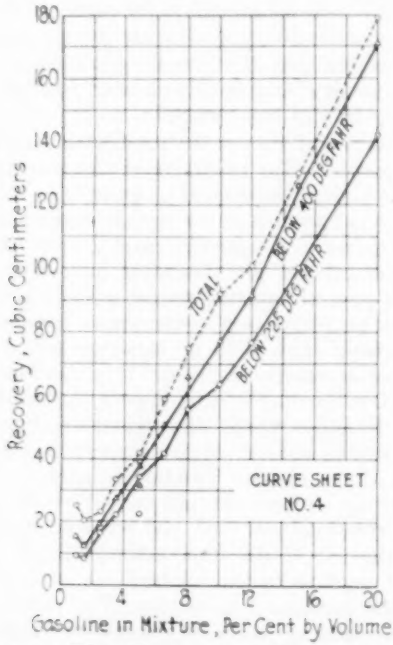

FIG. 10 RELATION OF TOTAL RECOVERIES TO RECOVERIES BELOW CERTAIN TEMPERATURES

(Graphical presentation of results given in Table 11. Mixtures containing absorption oil No. 1. This relation was determined from the plots of the distillations of the recoveries. See Curve Sheets 6 and 7 and Table 7.)

the former for recoveries from mixtures with absorption oil No. 1 and the latter for recoveries from mixtures with oil No. 2. The results have been plotted on Curve sheets 6, 7 and 8, Figs. 12, 13 and 14.

81 The recoveries from mixtures with absorption oil No. 2 represent more nearly ideal than ordinary test conditions. There was no variation in the characteristics of absorption oil No. 2. The oil was used as shipped. For each test a new sample was used and the oil from the still was discarded. In the case of absorption oil

No. 1 the oil could not be used as shipped since its over point was below 450 deg. fahr. Before absorption oil No. 1 was used the lighter fractions were distilled off. After the gasoline had been distilled over in the fractional distillation apparatus, the absorption oil left in the still was used a second time. To obtain complete data on oil No. 1 it was necessary to make a distillation similar to the distillation of the recoveries on a sample of each oil used. The data on these distillations are given in Table 8.

82 Tables 11 and 12 show the relation of total recoveries to the recoveries below certain temperatures. One temperature chosen was 400 deg. fahr. and the percentages of the recoveries below this temperature were read from the plots of the recovery distillations (Curve sheets 6, 7 and 8, Figs. 12, 13 and 14). It should

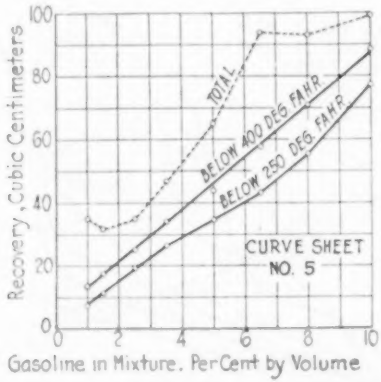

FIG. 11 RELATION OF TOTAL RECOVERIES TO RECOVERIES BELOW CERTAIN TEMPERATURES

(Graphical presentation of results given in Table 12. Recoveries from mixtures containing absorption oil No. 2. These relations were determined from data plotted on Curve Sheet No. 8. See Table 10.)

be understood that this is only an approximation of the fractions in the recovery below 400 deg. fahr. The apparatus used to distill the recoveries did not distill fractionally. When recoveries were distilled they were all treated similarly. So by taking the 400 deg. fahr. ordinate on the plot as a datum and reading the percentages over at that temperature a better comparison of recoveries was obtained. Percentages were read along a lower ordinate for both series of tests. The total recovery was then multiplied by these percentages and the results are tabulated in Tables 11 and 12. The data pertaining to absorption oil No. 1 are plotted on Curve sheet No. 4, Fig. 10, and those pertaining to oil No. 2 and plotted on Curve sheet No. 5, Fig. 11.

83 Curve sheets No. 4 and No. 5 show that the points of the recovery curves may be brought closer to a straight-line relation

by referring to some ordinate as datum. With the lower temperature ordinate (225 deg. or 250 deg.) as datum the results were not as satisfactory as with the higher 400 deg. ordinate. The recovery "below 400 deg." for absorption oil No. 2 plotted as a straight line. This shows that by treating the results of a fractional distillation properly the accuracy may be improved greatly. This is noticed especially in the treatment of the data from the 6½ per cent mixture (absorption oil No. 2). In this special case the distillation was allowed to run longer, and more of the heavier

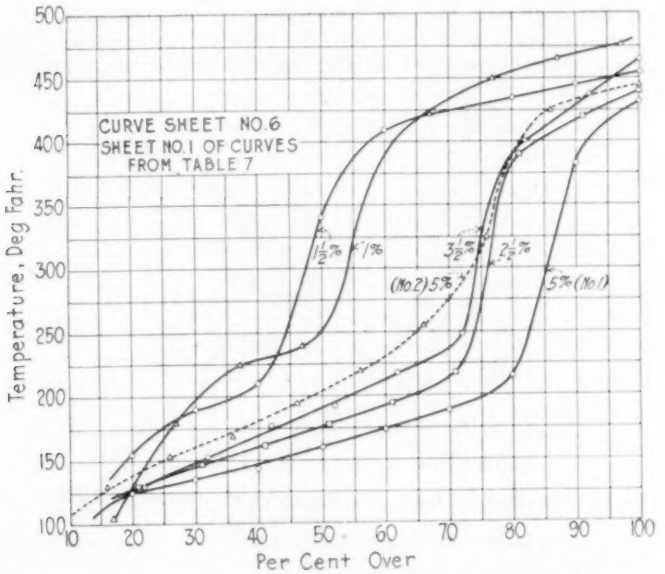

FIG. 12 DISTILLATIONS OF RECOVERIES BY SPECIAL APPARATUS

(Recoveries from mixtures of absorption oil No. 1 and 71 deg. B. gasoline. The basis of these curves is Table 7. See page 1236 for discussion of method of plotting, and Table 11 for data taken from these curves.)

fractions were distilled into the recovery. By the above treatment of results this recovery checks with the other recoveries.

84 This treatment of results takes extra time and would be objectionable when used for practical testing. Also, it would be difficult to set a standard of efficient operation to work to, since a slight variation in the properties of the absorption oil would make a large difference in the recovery. This last point is emphasized by the comparison of the "below 400 deg." curves on the two curve sheets. This curve is a straight line for the absorption oil No. 2 mixtures and these mixtures are identical except for the variation in gasoline content. The points of the "below 400 deg." curve for

absorption oil No. 1 do not fall as closely to a straight line as the points for oil No. 2. The point for the 1 per cent mixture is the weakest part of the curve for absorption oil No. 1. This is believed to have been caused by the difference in the samples of absorption oil No. 1 used for mixtures. Different test conditions may have partly affected the results.

85 The significant fact is that there is a great possibility of error when tests are made on mixtures having low percentages of gasoline. Comparative data are difficult to obtain when a slightly

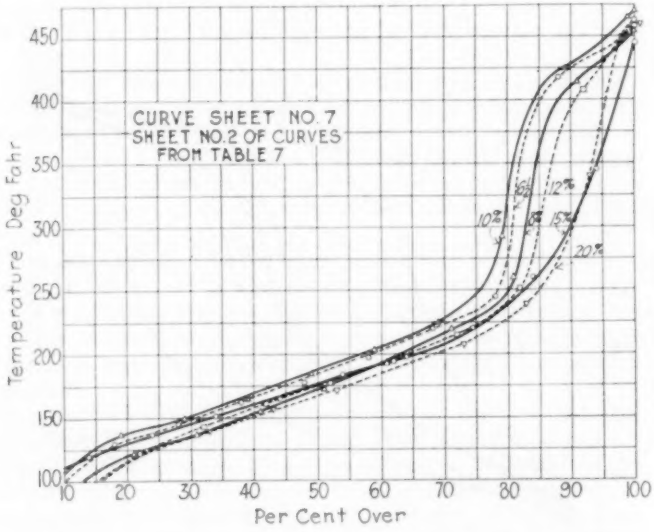

FIG. 13 DISTILLATIONS OF RECOVERIES BY SPECIAL APPARATUS
(See Par. 104 for method of plotting.)

different absorption oil is used in the mixtures or when test conditions are not well regulated.

86 In the discussion up to this point little has been said about the bearing of the results upon the test method used in the field. As has been stated, in the field the recovery is tested for specific gravity as the distillation in the test apparatus proceeds. When the specific gravity of the gasoline being produced by the plant is reached the volume of the recovery is noted and compared with the volume of the sample. This comparison gives the percentage of gasoline in the sample and that is considered the gasoline content of the absorption oil.

87 This procedure would give fairly consistent results for fat absorption oils, but it is doubtful whether accurate results could

be obtained from tests on lean oil since the recovery would be so small and the fractionating column so inefficient. Even with 12 in. of the column compactly filled with small jack chains, a large carry-over is found in the recovery. Curve sheets 6, 7 and 8 (Figs. 12, 13 and 14) show roughly the percentages of heavier fractions drawn over by the lighter vapors. They show that, if all other conditions were equal, the carry-over increases in percentage as the gasoline content of the sample decreases. The results show that consistent and comparative results may be obtained and relied upon only when the conditions are very good.

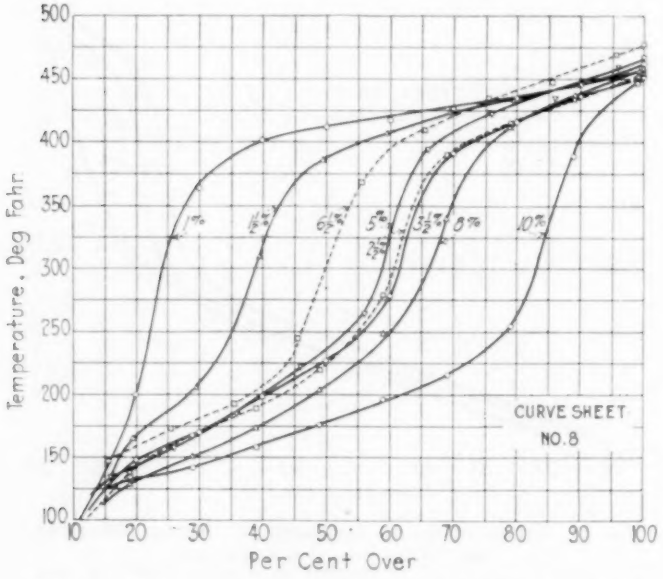

FIG. 14 DISTILLATIONS OF RECOVERIES BY SPECIAL APPARATUS

(Recoveries from mixtures of absorption oil No. 2 and 71 deg. B. gasoline. The basis for these curves is Table 10. See Table 12 for data taken from these curves.)

88 When the time taken in making a fractional distillation and the time taken in extra study is considered, the accuracy of the distillation method is out of proportion to the time required. The time element is one of the more important arguments against the distillation apparatus for determining the gasoline content. Another important point against the method is the large variation in specific gravity and volume of recovery with a slight variation in the characteristics of the absorption oil. This, however, may be disregarded if there is proof that the characteristics of the oil do not vary within certain limits.

PART V SUMMARY OF ADVANTAGES AND DISADVANTAGES

Distillation Method

89 *Advantages. a* The distillation method will give data on more than one of the physical characteristics of the recovery from an absorption oil sample. It will therefore give more information concerning the gasoline content of the absorption oil. By accurate treatment of the recovery, interesting facts may be discovered concerning the physical properties of the gasoline being produced by the plant. Accurate data are obtained only under very nearly ideal conditions; unless such conditions can be obtained this is not an advantage.

90 *b* For very lean oils the specific gravity of the recovery gives an indication of the gasoline content which can be compared without considerable error to the gasoline content of another oil. This advantage is obtained if the distillation apparatus is operated as it was in these tests and not as it is operated in practice. This specific gravity relation holds for very lean oils even if conditions are not ideal.

91 *c* An advantage of secondary importance lies in the size of sample tested by the fractional-distillation apparatus. A large and representative sample is used in this apparatus. However, the error due to a slight deviation in the characteristics of the sample is negligible when the magnitude of the other errors involved is considered.

92 *Distillation-Method Disadvantages. a* Perfect operating conditions are very difficult to obtain. Unless conditions are extremely constant during all the tests which are to be compared the results which are obtained cannot be satisfactorily compared.

93 *b* There is an exception to disadvantage *a*. This has been discussed under advantage *b*. That is, for lean oils the specific gravity of the recovery may give a fair indication of the gasoline content. The volume recovered is a rough indication of the gasoline content for fat oil. It is a disadvantage to have to rely upon the specific gravity for one range of gasoline content and the volume for another range, especially when the reliability of each indication decreases at the common percentages of gasoline content.

94 *c* Time spent in making a distillation test is out of proportion to the accuracy obtained. An accurate fractional distillation cannot be made in haste.

95 *d* Extra treatment of the recovery and data from the distillation is necessary to assure accuracy in comparison of results. This extra treatment requires additional time and apparatus.

Vapor-Pressure Method

96 *Advantages. a* If vapor pressure data will give a true, comparative indication of the gasoline content the vapor-pressure

method is very desirable because of the time saved. Any one of the three ways of making vapor-pressure determinations described in this paper takes comparatively little time after the apparatus is properly set up. Many vapor-pressure determinations may be made in the time taken for a distillation.

97 The accuracy of the vapor-pressure method may vary greatly according to the apparatus used and the conditions under which the tests are made. The results of this investigation show that under proper operating conditions consistent and comparative data may be obtained.

98 *b* Vapor pressure of the lean absorption oil is important when the operation of the absorber is considered. The amount of casinghead gasoline that an oil will absorb is dependent upon the difference between the vapor pressure of the oil entering and leaving the absorber and that of the partial pressure of the gasoline vapors of the natural gas entering and leaving. That is, if the lean oil enters the absorber with a high vapor pressure it will be impossible to properly strip the gas of its gasoline content.

99 *Disadvantages.* *a* A practical vapor-pressure apparatus such as apparatus No. 2 does not give absolute vapor pressure. This apparatus cannot be used except for comparative results. In order that good comparative results may be obtained some of the lighter fractions of gasoline are expelled from the sample when the air in the sample is removed. These lighter fractions cannot be kept from vaporizing when the air is being removed under vacuum unless the sample is held at a very low temperature. This would have to be done by means of liquid air or liquid ammonia. When an indication of gasoline content is the result desired it is not a disadvantage to have the very light fractions removed.

100 *b* Vapor-pressure apparatus No. 1 and No. 2 need quantities of clean and dry mercury. Although it is not difficult to purify mercury it takes time and extra apparatus. This is a secondary disadvantage that must be considered.

Comparison of the Vapor-Pressure Methods

101 Vapor-pressure apparatus No. 1 (cistern barometer) is very simple in operation and with ideal conditions it will give fairly consistent results. In practice the mercury is not boiled in the barometer tube. This leaves air between the mercury column and the walls of the tube. Conditions would be ideal if the amount of air carried to the top of the column was equal for each test and if that equal amount of air contained the same amount of water vapor each time. This air and the air contained in the sample cause a considerable error. If accurate results are desired from this apparatus it becomes necessary to boil the mercury in the tube and to treat the samples of oil under vacuum at a low temperature before injection. When the proper precautions are

taken the simplicity of operation is lost and the most important advantage is eliminated.

102 Vapor-pressure apparatus No. 2 (siphon barometer) is also very easily operated. Certain precautions, similar to those mentioned for apparatus No. 1, must be taken or the results when using such apparatus will not be comparative. Care must be taken that no air leaks into the vacuum leg of the barometer and after each determination the leg must be cleaned. With this apparatus and the procedure described in Pars. 41 to 44 the desired results were obtained.

103 The desirability of this investigation was suggested by Prof. W. R. Eckart of the Mechanical Engineering Department, Stanford University, who also gave such advice as was requested.

APPENDIX NO. 1 — EXPERIMENTAL DATA

EXPLANATION OF PLOTTING FIGS. 12, 13, AND 14, AND TABLES GIVING RESULTS OF EXPERIMENTS

104 Curve sheets 6, 7 and 8, Figs. 12, 13 and 14, and the corresponding tables (7 and 10) cannot be checked unless some explanation of the plotting is given. The apparatus used was special in that it has been superseded by more efficient apparatus for standard test distillations. It did not condense all of the lighter fractions of the recovery. The distillation loss was large and not very constant. For this reason the

TABLE 1 CISTERN BAROMETER — SECOND SERIES OF TESTS

Conditions: Jacket-water temperature constant at 60 deg. fahr. No attempt made to remove air from test samples. See data, Table 6A, for characteristics of absorption oil No. 1 and gasoline (same as used for all tests).

Percentage of gasoline in mixture	Date	Barometer, in. Hg	Room temp., deg. fahr.	Cistern-barometer readings: Initial, in.	Final, in.	Diff., in.	Oil on top, in.	Correction, in.	Corrected difference: In.	Mm.	Vacuum space, in. of tubing
1	8/17/23	30.09	78	30.10	28.57	1.53	0.7	0.041	1.489	37.8	3.4
2	8/17	30.09	79	30.08	28.155	1.925	0.75	0.044	1.881	47.7	2.2
3	8/17	30.12	72	30.115	28.27	1.845	1.0	0.059	1.786	46.3	4.1
4	8/17	30.10	75	30.09	27.995	2.095	0.8	0.047	2.048	52.0	2.9
5	8/16	30.08	78	30.035	27.825	2.21	0.98	0.058	2.152	54.6	3.0
6	8/18	30.11	72	30.105	27.39	2.715	1.3	0.077	2.638	66.9	3.1
8	8/18	30.11	74	30.11	27.25	2.86	1.0	0.059	2.80	71.1	4.4
10	8/18	30.12	74	30.115	26.64	3.475	1.2	0.071	3.40	86.3	4.1
				Check Readings							
1	8/20	30.06	72	30.09	28.715	1.375	0.8	0.047	1.328	33.7	2.4
2	8/20	30.08	72	30.075	28.40	1.675	1.0	0.059	1.616	41.0	3.1
8	8/18	30.11	74	30.105	27.15	2.955	1.1	0.065	2.890	73.4	3.8

TABLE 2 CISTERN BAROMETER — FIRST SERIES OF TESTS

Conditions: Jacket-water temperature constant at 60 deg. fahr. No attempt made to remove air from test mixtures. See data, Table 6A, for characteristics of absorption oil No. 1 and gasoline.

Percentage of gasoline in mixture	Date	Barometer, in. Hg	Room temp., deg. fahr.	Cistern-barometer readings: Initial, in.	Final, in.	Diff., in.	Oil on top, in.	Correction, in.	Corrected reading: In.	Mm.	Vacuum space, in. of tubing
0	8/1/23	30.01	74	29.99	28.62	1.37	0.8	0.047	1.32	33.5	...
1	8/3	30.08	70	30.065	28.88	1.185	0.6	0.035	1.15	29.2	2.65
2	8/3	30.07	72	30.035	28.51	1.525	0.7	0.041	1.484	37.8	2.8
3	8/6	30.02	70	30.015	28.06	1.955	1.0	0.059	1.896	48.1	...
4	8/4	30.12	68	30.045	28.225	1.82	0.9	0.053	1.767	44.8	3.2
5	8/4	30.10	74	30.02	27.47	2.55	1.0	0.059	2.49	63.3	3.0
7	8/4	30.10	71	30.075	27.525	2.55	0.95	0.056	2.494	63.4	4.0
10	8/4	30.09	73	30.05	26.995	3.055	1.05	0.062	2.99	76.0	5.0
15	8/6	30.02	70	30.005	26.005	4.0	0.85	0.05	3.95	100.5	4.8
20	8/8	30.03	71	30.01	24.435	5.585	1.5	0.089	5.5	139.5	5.7
				Check Readings							
0	8/14	30.06	78	30.055	28.965	1.09	1.0	0.059	1.03	26.2	2.7
1	8/8	30.06	71	30.04	28.72	1.32	0.9	0.053	1.267	32.2	2.8

data were plotted using 100 per cent over as a datum instead of 0 per cent. For the purpose of recording, 0 per cent had to be used as a datum. The curves as plotted indicate the distillation loss at the lower temperatures and not at the higher temperatures and therefore are more representative of the recoveries. The important data are relative to the end point of the recovery or the amount of heavy fractions in the recoveries. These distillations give good comparative data on the heavier fractions.

105 By investigating these curve sheets it may be seen that the end point and the percentage of heavier fractions in the oil are not constant. Even with careful regulation in running a fractional distillation, there will be a great variation in the end point and the percentage of light and heavy fractions in the recovery. With this variation in percentage of heavy fractions there will be a corresponding variation in the specific gravity of the recovery.

TABLE 3 SIPHON BAROMETER

Conditions: Jacket-water temperature (at time of reading) = 60 deg. fahr. Vacuum space constant at 70 mm. of tubing. Each mixture repeatedly put under vacuum until constant reading was obtained. See data, Table 6A, for characteristics of absorption oil No. 1 and gasoline.

Percentage of Gasoline in mixture	Date	Barometer, in. Hg	Room temp., deg. fahr.	Siphon-barometer readings				Correction, mm.	Corrected reading, mm.
				Initial, mm.	Final, mm.	Diff., mm.	Oil on top, mm.		
0	8/20/23	30.08	75	756	749.5	5.5	45.6	2.7	2.8
1	8/23	30.05	72	756	744	12	58.5	3.46	8.5
2	8/21	30.06	74	756	740	16	56	3.3	12.7
3	8/22	30.12	72.5	757	736.5	20.5	52	3.1	17.4
4	8/21	30.06	80	757	733	24	51	3.0	21.0
5	8/22	30.05	78	756.5	728	28.5	70	4.1	24.4
6	8/23	30.04	74	757	721.5	35.5	53	3.2	32.3
8	8/22	30.05	74	755.5	711	44	52	3.1	40.9
10	8/22	30.05	73	757	701.5	55	51	3.0	52
15	8/24	30.04	76	756	680.5	76.5	58	3.5	73
20	8/23	30.00	76	757	659	98	58	3.5	94.5

TABLE 4 SIPHON BAROMETER

Conditions: Sample was put under vacuum and allowed to stand until there was no bubbling or frothing before a reading was made. Jacket-water temperature (at time of reading) = 60 deg. fahr. Vacuum space constant at 70 mm. of tubing. See Table 6A for data on oil and gasoline used.

Percentage of Gasoline in mixture	Date	Barometer, in. Hg	Room temp., deg. fahr.	Siphon-barometer readings				Correction, mm.	Corrected reading, mm.
				Initial, mm.	Final, mm.	Diff., mm.	Oil on top, mm.		
1	8/17/23	30.09	78	757.0	706.5	50.5	48	2.8	47.7
2	8/17	30.09	79	754.5	711	43	41	2.4	40.6
3	8/17	30.12	72	756.5	705	51	51	3.0	48
4	8/17	30.10	75	756	699.5	56.5	44	2.6	53.9
5	8/16	30.08	78	756	690.5	65.5	45	2.5	63
6	8/18	30.11	72	757.5	684.5	73	48	2.8	70.2
8	8/18	30.11	74	755	666	89	25	2.5	87.5
10	8/18	30.12	74	757	661	96	49.5	2.9	93.1
				Check Readings					
1	8/20	30.08	72	756.5	702	54.5	48	2.8	51.7
2	8/20	30.08	72	756	692	53.5	51	3.0	50.5
8	8/18	30.11	74	755	671	84	51	3.0	81

TABLE 5 SIPHON BAROMETER

Conditions: Readings were taken hastily before vaporization was complete and before bubbling had ceased. Jacket-water temperature constant at 60. deg. fahr. Vacuum space constant at 70 mm. of tubing. See Table 6A for data on oil and gasoline used.

Percentage of gasoline in mixture	Date	Barometer, in. Hg	Room temp., deg. fahr.	Siphon-barometer readings				Correction, mm.	Corrected reading, mm.
				Initial, mm.	Final, mm.	Diff., mm.	Oil on top, mm.		
0	8/1/23	30.01	74	755.5	709.5	46	..	...	46
1	8/3	30.08	76	755	728	27	51	3.0	24
2½	8/3	30.07	72	757.5	696	61.5	58	3.4	58.1
3	8/6	30.02	70	756	687	69	53	3.1	65.9
4	8/4	30.12	68	758	707	51	46	2.7	48.3
5	8/4	30.10	74	759.5	682.5	37	35.5	2.1	74.9
7	8/4	30.10	71	757.5	667	90.5	53	3.1	87.4
10	8/4	30.09	73	757.5	653	104.5	48	2.8	101.7
15	8/6	30.02	70	756.5	612.5	144	58	3.4	140.6
20	8/8	30.03	71	757.5	577	180.5	51	3.0	177.5
				Check Readings					
0	8/14	30.06	78	760.5	699.5	61	48	2.8	58.2
1	8/8	30.05	71	756	689.5	67	58	3.4	63.6

TABLE 6 RESULTS FROM TESTS WITH FRACTIONAL-DISTILLATION APPARATUS

Conditions: See Pars. 51 to 55 for the procedure. Additional data on recoveries are given in Table 7. Absorption oil No. 1 was varied. See Table 8. Gasoline in these mixtures is described in Table 6A.

Percentage of gasoline in mixture	Date	Barometer, in. Hg	Temperature, deg. fahr.			Recoveries			Recovery, sp. gravity
			Room	Flask	Column	Graduate, cu. cm.	Cotton, mg.	Total, cu. cm.	
1	7/25/23	30.08	73	...	462	25	265	25.3	0.8082 at 18° C.
1½	8/16	30.11	72	528	450	20.5	210	20.5	0.7787 at 16° C.
2½	8/13	30.10	78	522	450	24.5	174	24.5	0.7304 at 14° C.
3	8/14	30.09	70	521	450	33.5	263	34	0.7328 at 15° C.
5	8/14	30.05	76	526	450	42	184	42	0.7142 at 18° C.
5	7/22	30.04	74	...	460	39	...	39	0.7258 at 13° C.
6½	8/5	30.08	74	524	454	59	150	59	0.7200 at 14° C.
8	8/15	30.06	79	523	452	75	246	75	0.7132 at 17° C.
10	7/20	30.05	72	...	460	92	...	92	0.7243 at 13° C.
12	8/16	30.11	74	522	450	101	155	101	0.6960 at 16° C.
15	7/23	30.06	74	...	455	130	...	130	0.7102 at 13° C.
20	7/24	30.09	72	...	462	179	...	179	0.7079 at 15° C.

TABLE 6A SPECIFIC GRAVITY AND DISTILLATIONS OF ABSORPTION OIL AND GASOLINE USED FOR VAPOR-PRESSURE DETERMINATIONS

	Sp. gravity	Over point	Percentages 10	20	30	40	50	60	70	80	90	End point
							Deg. fahr.					
Absorption oil No. 1 (A)[1]	0.8526 at 22½° C.	465	486	499	510	520	531	546	562	581	612	620
Absorption oil No. 1 (B)[2]	0.8568 at 21° C.	465	493	515	523	534	544	553	572	591	618	630
Gasoline used for all mixtures	0.6958 at 22° C.	90±	122	140	160	178	193	220	271	460	86%-end point 479	

[1] Absorption oil No. 1(A) used to obtain results in Tables 1, 3, and 4.
[2] Absorption oil No. 1(B) used to obtain results in Tables 2 and 5.

TABLE 7 DISTILLATION OF RECOVERIES

Recoveries from mixtures containing absorption oil No. 1. See Table 6 for data on distillations to obtain these recoveries. Recovery distillations were made in the special apparatus and in accordance with the procedure described in Pars. 106 to 109.

Gasoline content in the mixture, percentage by vol.	Date	Recovery used, cu. cm.	Recovery, specific gravity	Over point	Percentages 10	20	30	40	50	60	70	80	End point
									Deg. fahr.				
1	7/25	25	0.8082 at 18° C.	105	180	225	240	...	420	450	466	476	482 83 per cent
1½	8/16	20½	0.7787 at 16° C.	90	155	190	210	340	410	...	435	446	455 90 per cent
2½	8/13	23	0.7304 at 14° C.	85	130	148	162	178	195	218	390	420	440 89 per cent
3½	8/14	33½	0.7328 at 15° C.	85	130	152	176	193	219	249	400	438	465 88 per cent
5	8/14	42	0.7142 at 18° C.	85	125	135	144	160	173	188	215	382	432 90 per cent
5	7/22	39	0.7258 at 13° C.	95	130	154	169	195	220	255	325	425	445 94 per cent 90% 92%
6½	8/15	59	0.7200 at 14° C.	90	130	145	162	178	197	220	246	418	448 458
8	8/15	75	0.7132 at 17° C.	90	122	138	156	174	194	220	260	415	456 89 per cent 90% 91%
10	7/20	92	0.7243 at 13° C.	100	125	150	165	184	203	225	287	425	465 470
12	8/16	101	0.6990 at 16° C.	95	123	143	161	178	195	215	251	408	462 88 per cent 90% 96%
15	7/23	130	0.7102 at 13° C.	100	121	137	152	165	184	198	220	260	345 445 85% 87%
20	7/24	179	0.7079 at 15° C.	100	125	140	158	172	...	208	242	340	450 460

TABLE 8 DISTILLATIONS OF "LEAN" ABSORPTION OILS NO. 1

These oils were used in making the mixtures tested by the fractional-distillation apparatus. See Table 6 for results of tests on the mixtures. The distillations below were made in the special apparatus and in accordance with the procedure described in Pars. 106 to 109.

Mixture in which the oil was used, per cent	Date	Amount distilled, cu. cm.	Specific gravity	Percentages: Over point	10	20	30	40	50	60	70	80	90	End point
				Deg. fahr.										
1	7/25/23	50	0.8560 at 23° C.	475	501	511	521	529	539	550	...	595	609	615 97 per cent
1½	8/16	50	0.8535 at 22° C.	464	489	504	513	520	533	545	560	577	606	634 98 per cent
2½	8/13	50	0.8455 at 27° C.	460	475	495	504	516	536	546	557	579	608	624 96 per cent
3½	8/13	50	0.8455 at 27° C.	460	475	495	504	516	536	546	557	579	608	624 96 per cent
5	8/14	50	0.8465 at 24½° C.	460	490	502	509	522	537	550	565	592	608	620 97 per cent
5	7/22	..	0.8563 at 22° C.	480	510	520	529	537	546	556	...	590	630	648 96 per cent
6½	8/15	50	0.8537 at 22° C.	465	484	500	510	519	530	540	556	563	610	620 95 per cent
8	8/15	50	0.8528 at 23½° C.	462	486	500	512	522	535	545	565	579	616	630 98½ per cent
10	7/20	..	0.8524 at 24° C.	440	469	480	....	505	517	529	...	560	582	600 98½ per cent
12	8/16	50	0.8526 at 22½° C.	465	486	499	510	520	531	546	562	581	612	620 99 per cent
15	7/23	..	0.8564 at 22° C.	460	...	515	524	536	544	552	560	...	602	689 98 per cent
20	7/24	50	0.8566 at 22° C.	485	497	...	519	531	542	...	565	581	620	638 99 per cent

TABLE 9 RESULTS OF TESTS WITH FRACTIONAL-DISTILLATION APPARATUS

Gasoline content, percentage by vol.	Date	Barometer, in. Hg	Temperatures, deg. fahr.: Room	Flask	Column	Recovery: Graduate, cu. cm.	Cotton, mg.	Total, cu. cm.	Specific gravity
1	8/27/23	30.02	74	516	449.5	36.5	284	37	0.8079 at 15° C.
1½	8/27	29.96	81.5	516	449	32	175	32	0.7850 at 19° C.
2½	8/26	30.02	77	513	449	35	245	35	0.7570 at 13° C.
3½	8/26	29.97	81.5	512	449	47	283	47	0.7575 at 18° C.
5	8/25	29.97	78	514	453	67	185	67	0.7510 at 18° C.
6½	8/27	29.95	84.5	518	458	94	...	94	0.7580 at 17° C.
8	8/28	29.95	75.5	514	451	93	282	93	0.7426 at 14° C.
10	8/25	29.98	76	512	449.5	99	113	99	0.7195 at 13½° C.

CONDITIONS:

See Pars. 51 to 55 for the procedure. Additional data on recoveries are given in Table 10. Absorption oil in these mixtures did not vary (see Table 10A), nor did the gasoline (see Table 6A). Absorption oil No. 2 (used for the mixtures of this table) did not contain light fractions. Its over point was 460 deg. fahr. when distilled in the special apparatus.

TABLE 10 DISTILLATIONS OF RECOVERIES

Recoveries from mixtures containing absorption oil No. 2. See Table 9 for data on distillations to obtain the recoveries. Recovery distillations were made in the special apparatus and in accordance with the procedure described in Pars. 106 to 109.

Gasoline content in oil, percentage by vol.	Date	Recovery used, cu. cm.	Specific gravity	Over point	Percentages 10	20	30	40	50	60	70	80	90	End point
				Deg. fahr.										
1	8/27	35	0.8079 at 15° C.	95	200	364	401	412	418	427	435	446	462	462 90 per cent
1½	8/27	32	0.7850 at 19° C.	95	160	206	310	386	407	424	434	446	457	462 90½ per cent
2½	8/26	35	0.7570 at 13° C.	85	138	168	189	220	281	390	415	435	452	455 91 per cent
3½	8/26	47	0.7575 at 18° C.	85	148	176	199	227	278	391	417	437	452	452 90 per cent
5	8/25	65	0.7510 at 18° C.	85	136	159	186	222	264	394	422	436	466	468 94 per cent
6½	8/27	94	0.7585 at 17° C.	95	147	174	193	245	369	450	436	449	470	482 94½ per cent
8	8/28	93	0.7426 at 14° C.	100	129	151	174	204	249	340	412	437	459	460 91 per cent
10	8/25	99	0.7196 at 13½° C.	85	137	142	159	176	142	216	254	389	448	451 91 per cent

TABLE 10A DISTILLATION OF THE "LEAN" ABSORPTION OIL NO. 2

Apparatus and procedure same as for Table 10. This represents the oil as shipped and as used for the mixtures of Table 9.

Date	Amount distilled,	Specific gravity	Over point	Percentages 10	20	30	40	50	60	70	80	90	End point
			Deg. fahr.										
Aug. 25, 1923	100 cu. cm.	0.8460 at 23° C.	460	477	491	499	510	520	531	545	564	595	652	99 per cent

TABLES 11 AND 12 SHOWING THE RELATION OF TOTAL RECOVERY TO THE RECOVERY BELOW CERTAIN TEMPERATURES

TABLE 11

Per cent readings were obtained from distillation curves (Figs. 12 and 13)

Percentage of gasoline in mixture	Percentage of recovery below 400° F., datum	Percentage of recovery below 225° F., datum	Total recovery from mixture, cu. cm.	Recovery below 400° F., cu. cm.	Recovery below 225° F., cu. cm.
1	61.5	37	25	15.4	9.25
1½	58	42	20½	11.9	8.6
2½	84.5	72	23	19.4	16.6
3½	82	65.5	33.5	27.4	22
5 (No. 1)	91.5	81	42	38.4	34
5 (No. 2)	81	58	39	31.6	22.2
6½	85	70.5	59	50.2	41.6
8	88	74	75	66	55.5
10	84	69	92	77.4	63.5
12	90.5	76	101	91.5	76.8
15	97	77	130	126	100
20	95.5	79	179	171	142

TABLE 12

Per cent readings were obtained from distillation curves on Fig. 14

Percentage of gasoline in mixture	Percentage of recovery below 400° F.	Percentage of recovery below 250° F.	Total recovery, cu. cm.	Recovery below 400° F., cu. cm.	Recovery below 250° F., cu. cm.
1	39.5	22	35	13.8	7.7
1½	56	35.5	32	17.9	11.9
2½	72	55	35	25.2	19.2
3½	73	56	47	34.3	26.3
5	67.5	53.5	65	43.9	34.8
6½	61.5	46	94	57.7	43.2
8	76	59.5	93	70.7	55.4
10	89.5	78	99	88.5	77.1

TABLE 13 COMPARISON OF ABSORPTION OILS

	Bureau of Mines, Bulletin No. 176, "a good absorption oil"	Absorption oil No. 1, light fractions off	Absorption oil No. 2, as shipped
Flash point,[1] deg. fahr.	247	228 [1]	231 [1]
Fire test, deg. fahr...	262	245 [2]	252 [2]
Specific gravity, deg. B.	37	34	35.5
Viscosity	0.0620 abs. at 20° C.	0.0559 abs. at 70° F.	0.0499 abs. at 71° F.
Over point, deg. fahr..	450	460	460
End point, deg. fahr..	695	630	652

[1] Pensky-Martin closed-cup flash tester.
[2] Cleveland open-cup tester.

DISTILLATIONS

	Over point	Percentages 10	20	30	40	50	60	70	80	90	End point
	Deg. fahr.										
Bulletin No. 176, "a good absorption oil"	450	528	540	556	565	572	572	595	608	633	695
Absorption oil No. 1, "with light fractions"	340	445	460	480	495	506	520	535	546	580	615
Absorption oil No. 1, "light fractions off" [1]	460	475	495	504	516	536	546	557	579	608	624
Absorption oil No. 2, "as shipped"	460	477	491	499	510	522	531	545	564	595	652

[1] Used for the 2½ and 3½ per cent mixtures. Taken as representative of all mixtures.

APPENDIX NO. 2 — APPARATUS FOR DISTILLATION OF RECOVERIES

106 The distillation apparatus shown in Fig. 15 has been superseded as a standard apparatus for testing petroleum and its products. This apparatus served the purpose of these tests very well, but the results cannot be compared to those obtained with another apparatus which is not identical.

107 This apparatus served a double purpose. First, it gave comparative data on the recoveries from the fractional-distillation apparatus; second, it showed by comparative data the uniformity of absorption oil No. 1 which was used in making the test mixtures of Table 6. In addition it gave approximate information of the distillation of the gasoline used for making mixtures.

108 The apparatus consisted of a 250-cu. cm. glass distilling flask, a water-jacket condenser tube and a 100-cu. cm. graduate which was

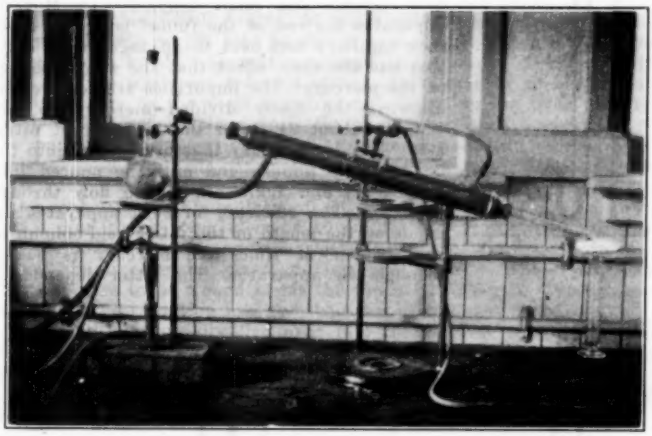

FIG. 15 APPARATUS FOR DISTILLATION OF RECOVERIES

the receiver. The bulb of a 680-deg. fahr. thermometer was placed in the neck of the flask about ½ in. below the connection of the arm and the neck. The bulb was wrapped with cotton so that there would be no possibility of error due to superheating. A piece of bent glass tubing was attached to the end of the condenser tube. This tube and the piece of asbestos sheet covering the receiver reduced the distillation loss slightly. The receiver was placed in a glass jar which was filled with cracked ice. This did not prove very satisfactory. The ice in the jar made it difficult to make accurate readings and the distillation loss was very large even with this precaution. The loss was chiefly due to the inefficiency of the water-jacketed condenser.

109 When the recovery or the sample of oil to be tested had been poured into the flask and all the connections had been made tight, the heat from the burner was applied cautiously. The distillation was carried on at the rate of one drop per second. The temperatures were read at the over point and for every 10 per cent of the total recovery over thereafter until the end point was reached. The procedure was the same for the testing of the absorption oils except that the receiver was not cooled by an ice jacket.

APPENDIX NO. 3 — PURIFICATION OF MERCURY.

METHOD NO. 1

110 Method No. 1 was used for purifying the mercury used in vapor-pressure apparatus No. 1 and apparatus No. 2. This method is similar to Lothar Mayer's method[1] in principle. The apparatus, shown in Fig. 16, is the one used for the tests of this investigation. It is a modification of Mayer's apparatus.

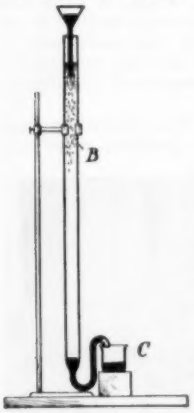

FIG. 16

111 The apparatus is set up as follows: The bottom of the tube *B* in the figure is first filled with impure mercury and a 10 per cent nitric acid solution is added. Mercury is then poured through the funnel *A*. The end of the funnel is covered with several layers of silk cloth which are tightly bound to the funnel tube. The mercury filters through the cloth and is divided into fine streams and small droplets. In Mayer's apparatus the end of the funnel is drawn down to a capillary and bent to an angle of 60 deg.; this has the same effect that the silk cloth has on the mercury. The impurities are removed by allowing the finely divided mercury to run through about 2½ ft. of the 10 per cent nitric acid. The dry mercury that first passes into the beaker *C* is impure and must be poured into the funnel and again allowed to flow through the acid. The efficiency of such apparatus depends on the length of the nitric acid column or the number of times the mercury is passed through the apparatus. The latter depends, of

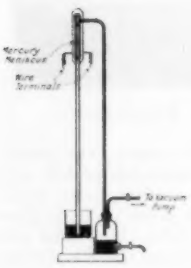

FIG. 17

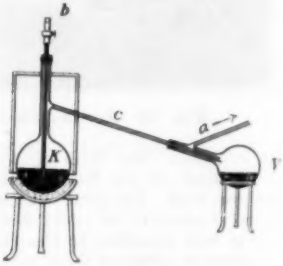

FIG. 18

FIG. 16 PURIFICATION OF MERCURY — METHOD NO. 1

FIG. 17 PURIFICATION OF MERCURY — METHOD NO. 2
(Copy of a drawing by E. F. Demond.)

FIG. 18 PURIFICATION OF MERCURY — METHOD NO. 3
(Copy of Fig. 109, Treadwell and Hall's Analytical Chemistry.)

[1] *Zeit. anal. Chem.*, vol. 2 (1863).

course, on the quantity and nature of the impurities. In the vapor-pressure determinations of this investigation oil was the chief impurity, and it was removed from mercury without much difficulty. By passing the mercury through the apparatus once a fairly pure mercury was obtained.

Method No. 2

112 Method No. 2 was not used for the purposes of these tests. However, it is a practical, exact, and convenient method for purification of small amounts of mercury. If any of the vapor-pressure methods are to be used continually for determinations, then such a method of cleaning mercury would be warranted.

113 The method is by distillation. In principle it is the same as that introduced by L. Dunoyer.[1] One modification has been made by E. F. Demond; the still is made of steel instead of glass. A sketch of E. F. Demond's still is shown in Fig. 17. The still is a cistern barometer with a heating element, surrounding the upper end of the barometer tube, which heats and vaporizes the top mercury of the column. A tube which acts partly as a condenser conveys the vapor to a vacuum bottle where it condenses and collects. The barometer tube and the condenser tube, as stated above, may be of either glass or steel. Due to the steel the first quantities coming over into the receiver will not be clean until the tubing has been washed out by the mercury. The receiving bottle has a connection to a vacuum pump which is used to fill the barometer tube by creating the initial vacuum in the tubes and bottle. The bottle also has a stopcock at the bottom which makes it possible to draw off mercury when the still is shut down.

114 For the heating element 10 ft. of No. 27 chromel resistance wire is sufficient. The wire is wound about the barometer tube as shown. Coats of water glass and layers of asbestos permit the total 10 ft. of wire to be wound about the top of the column. The asbestos cuts down the radiation loss and insulates at the same time.

115 The quantity distilled over into the receiver in a given time is not great. However, the still is reliable and it is safe to let it run for hours at a time without watching it.

116 The mercury should be cleaned by the dilute nitric acid method before it is distilled in this apparatus.

Method No. 3

117 The third method of purifying mercury is also by distillation. For the purposes of the tests of this investigation there was no great need to have absolutely clean mercury. Hulett's apparatus, shown in Fig. 18, may be used for distilling mercury that is to be used for calibration purposes.

118 The following is a description of the method by Treadwell:[2] The mercury is placed in the long-necked flask *K* which is connected with the receiver *V*. The flask is covered with a hood of asbestos paper and heated on the sand bath. Through the arm *a*, the receiver is connected with a suction pump and at *b* a slow current of nitrogen (or carbon dioxide) which has been dried by passing over calcium chloride, is introduced into the flask through the long glass tube that ends in a capillary. The distillation is regulated so that the mercury condenses in the glass arm *c*, where it leaves the hood of asbestos paper. About from 150 to 200 cu. cm. of mercury can be distilled in an hour

[1] L. Dunoyer: *Annales de Chemie de Physique*, vol. 26, pp. 419-432.
[2] Treadwell and Hall, Analytical Chemistry, vol. 2, pp. 747-748.

with this apparatus. Frequently, especially when the nitrogen used contains a little oxygen, the distilled mercury is covered with a thin coating of oxide. This may be removed by filtration. To filter mercury, the point of the paper filter is perforated several times with a needle the filter placed in a funnel and the mercury poured through. The pure metal runs through the holes in the paper while the impurity remains behind.

No. 1993

LARGE STEAM TURBINES

By Francis Hodgkinson,[1] Philadelphia, Pa.

Member of the Society

UNTIL now, I have confined myself to past and existing practice in turbine design.[2] My remarks would not be complete did I not refer to the present trend of design. It is well known that the steam turbine of large size, under reasonably favorable conditions, has reached a high degree of efficiency. Eighty per cent or more of what is theoretically possible from the expansion of the steam is today available at the shaft of the turbine. I feel, however, that the turbine designer has by no means reached the end of his tether and that there remains much that may be done. Certainly greater reliabiiity must be secured, and there still remains a modicum of attainment in efficiency. The percentage is small, it is true, but remember that in a base-load station of, say, 100,000 kw. capacity with 75 per cent load factor, a 1 per cent real gain in steam consumption (not necessarily that in a contract warranty) represents a capitalization for fuel alone of $175,000, based on a fifteen-year life, 6 per cent interest, and $4.50 per ton cost of fuel. That refers to the prime mover alone, but I think it is still true, though to a less degree than formerly, that it is not the efficiency of the prime mover that spells low cost of power. Highly efficient prime movers contribute little as compared with eternal vigilance in connection with boiler-house problems and matters of heat balance, feed heating, etc., in which subjects, however, great progress has been made of late.

2 Undoubtedly every designer, purchaser, or operator of a prime mover must fully subscribe to the doctrine of "Reliability first, economy second." Obviously some modicum of present efficiency had best be sacrificed if a greater degree of reliability cannot be secured in any other way, and there are those, indeed, who hold that this would be a wise course to pursue today. I venture

[1] Chief Engineer, Westinghouse Elec. & Mfg. Co., South Philadelphia Works.

[2] The author refers to a résumé of a paper which he presented at the World Power Conference, and which dealt with types of turbines and features of design. See *Mechanical Engineering*, vol. 46 (1924), p. 628.

Abridgment of an address delivered at the Second Annual Power Meeting, Chicago, Ill., Jan. 14 and 15, 1925, of the Chicago Section of The American Society of Mechanical Engineers.

to say that it is not an unusual experience for the designer of prime movers to find in his dealings with purchasing engineers that the latter have a dual personality. The designer, to begin with, is coerced to produce a machine of the highest possible economy, and to attain this he has sometimes subscribed to details perhaps not altogether well tried. With the best intentions and with full recognition of the doctrine of "Reliability first, economy second," he runs near the danger line, and this, combined with some unfortunate operating circumstances, causes outages in the operation of this prime mover. It is then he discovers the dual personality of the purchasing engineer, when he is asked what the purchasing engineer cares about one or two per cent of efficiency as compared with reliability. As a matter of fact, the purchasing engineer is correct in both phases of this dual personality.

3 Frankly, I personally would be sorry if designers were to depreciate the efficiency of their machines. They must, and can, secure increased reliability without this, for I hold that greater reliability is to be secured at the same time as a moderate gain in efficiency. Of course, high efficiency is to be secured with turbines designed for low leaving loss and, as I think it can be shown, with many stages, and while elaboration in design affects cost, I am not losing sight of fixed charges in my thoughts on these matters.

4 It seems to me that turbine practice in Europe is in a transitory stage, and that the changes are well worthy of study at our hands. European engineers are reaching out for higher efficiencies, and one must consider in the doing of it to what degree reliability may be imperiled and to what degree we on this side may profit thereby. Are their standards of reliability lower than ours? I can't answer that question. My impression is that utility companies carry more spare parts there than here, and do not expect the manufacturer to carry an insurance policy for them in this regard.

Tendencies in Design

5 I am sure all of us will recognize that able designing is not limited to this side of the Atlantic. This is amply shown by the purchase by one of the greatest public-utility companies of this city, and of the world, of a large and important machine — already referred to — a conception of a distinguished British engineer.

6 Some few years ago we were told of high efficiencies being at least guaranteed with high-velocity high-pressure elements, resulting in turbines of large capacities at given speeds with few stages — in instances as low as eight single-row stages — these being in some cases entirely of impulse type, and in others, combinations of impulse and reaction types. In spite of some recorded tests to the contrary, results with this type of machine have been disappointing, and designers, led by the classic experiments carried out by the Beama on nozzles, have come to the reali-

zation, long felt by at least some engineers in this country, that the best economy is to be secured with high-pressure elements running at low speeds. This is obviously so for reasons other than the somewhat abstruse ones indicated by the Beama tests, for, with a given steam flow, larger nozzle and blade passages may be employed, involving lesser leakage ratios and lower disk friction — all these things conducing to higher efficiency. The development of this thought — of a turbine of many stages in the high-pressure zone — has led many European designers to turbines of the tandem-compound principle, at least for their machines of maximum capacity at given speeds, in order to secure greater mechanical rigidity of the small shaft and many stages of the high-pressure portion, and to maintain fine clearances.

7 We have heard a good deal lately about the activities of the Erste-Brünner Maschinenfabrik and their exploitation of machines of a design based on the thought of high-pressure elements operated at low velocity. Remarkable results were secured with a 2300-kw. non-condensing machine adapted for service where exhaust steam at some pressure above atmospheric was needed for some industrial process. Authentic tests by Dr. Stodola showed an overall efficiency of the turbine alone of 82.4 per cent. It is obvious that such a turbine compounded with low-pressure elements would be a step in advance for a complete condensing power-house unit. The Erste-Brünner Company has now secured a number of licensees, particularly in Germany, and at least one in Holland, who are offering machines of this type, in fact, going apparently to the extreme in the matter of tandem-compounded cylinders. In one case a 50,000-kw. machine was lately offered in four tandem cylinders — one high pressure, one intermediate, and two single-flow low-pressure elements to operate at 1500 r.p.m. It is to operate with a pressure of 357 lb. per sq. in., 660 deg. fahr. total temperature, and 29 in. vacuum, and a steam consumption of 8.11 lb. per kw-hr. is expected.

8 At the World Power Conference, C. F. Stork, of the Gebrüder-Stork Co., of Holland, in referring to their adoption of the Erste-Brünner idea, says in this connection:

> The keen competition has resulted in a struggle of the various turbine builders to show low steam-consumption figures, and it can hardly be denied that every single one of the designs that has been applied in order to obtain these figures has been made at the expense of the safety of working. This is the reason why, in the course of 1923, it was decided to design and build a turbine on entirely new principles.

This company was building a machine — perhaps now already built — which was described at the World Power Conference, of 16,000 kw., to operate with 456 lb. pressure, 750 deg. fahr. temperature, and 69 deg. fahr. temperature of cooling water, on which 8.44 lb. per kw-hr. has been guaranteed, which corresponds to an overall efficiency of 86 per cent for the turbine alone. I do not doubt that designers have been led to this extreme multiplicity

of cylinders for the reasons stated by Mr. Stork — for the sake of increased reliability, securing rigidity with small-diameter shafts, and to enable them to operate with the interstage labyrinths with minimum clearance. I have been told, but cannot vouch for the truth, that axial clearances in these turbines between blades and nozzles approximate 0.005 in. Fine clearances at this point are of course conducive to efficiency. It must not be lost sight of that there are losses incidental to taking out steam from one cylinder by means of a pipe and admitting it to another. Notwithstanding this, however, excellent results will no doubt be secured with machines built on these lines, and a number of manufacturers, licensees and otherwise, are subscribing to this principle and their later important machines are being arranged as tandem-compound machines.

9 The adoption of the one-speed coupled tandem turbine is a reversion to the practice of both Parsons and Westinghouse of 25 years ago. With the relatively low rotative speeds of that day, I doubt whether much higher efficiency could have been secured in any other way and the machines built, though costly, were about as reliable as any today. Conditions, however, have changed in the respect of higher vacuum, and more particularly, higher speed of generators, as I have previously stated. It would seem, however, that the British have somewhat outstripped us in generator speeds inasmuch as 20,000-kw. single generators at 3000 r.p.m. have been constructed. One must not be misled, however, for there are differences in the practices of the two countries in the respect of relation of field strength to armature strength. The inherent voltage regulation is wider in their practice than in ours.

10 Of course, the principles of compounded turbines that I have been talking about, whether one-speed tandem-coupled, two-speed tandem-uncoupled, or two-speed cross-compounds, are applicable to impulse and reaction turbines alike. By such means higher efficiencies may be secured, I believe from 6 to 7 per cent, if appropriate low blade speeds are selected for the high-pressure elements and the low-pressure blades are designed for low enough leaving loss and between these there are enough stages to provide appropriate velocity ratios. The reliability will be increased to a greater or lesser degree according to the particular construction employed.

11 I need hardly say to many of you — and it is clear from the foregoing — that I personally have for many years advocated the cross-compounding of large turbines, for two particular reasons, which have been stated in part before. The one that by operating the low-pressure element at relatively low speed, any last-row blade area can be employed, and hence any degree of low terminal loss secured. Simple blade constructions, involving any degree of low stress desired, may be employed. The high-pressure element of

course will operate at a higher synchronous speed selected to give the highest efficiency with the steam volumes involved. The second reason for all the above is that there is added a real modicum of reliability by virtue of the greater simplicity and the lesser size of the two independent structures, and the lesser temperature range and heat drop to be reckoned with.

12 The large-capacity high-speed turbine generator of today is no small problem of design, and it may be greatly relieved by the two-speed-compound-turbine principle.

13 In the case of single-speed turbines (tandem-compound or single-cylinder) there seems to be a tendency in Europe, which I think should be followed here, of purchaser and manufacturer alike concentrating on and employing the maximum-capacity turbine at each of the available synchronous speeds, their best machines being built only in a limited number of sizes. Do not misunderstand me in this. I do not refer to machines where high capacity is secured at the expense of increased terminal loss, but to machines with design constants based on appropriate leaving losses, velocities of elements, and velocity ratios. At any given speed it is the least costly per kilowatt if similar design constants are employed, unless great cost is incurred in materials and construction for the purpose of securing a high rotative speed difficult of attainment. It is therefore the machine on which the most money can be spent in the minor details that tends to reliability and efficiency.

14 Other causes besides generator design have contributed to the reduction of steam consumption of all types of turbines. The higher speed permits reduced leakage and higher design constants such as velocity ratio without as much compromise for mechanical reasons. Higher vacuum has brought about increased cost of the extreme low-pressure portion of the turbine. Higher pressures and superheats have contributed greatly to better performance, but have added to intrinsic costs not only on account of the greater number of turbine elements to accommodate the increased heat drop but also because of more costly materials due to the higher temperatures. Every step in performance has resulted in higher intrinsic value, except where an increase of r.p.m. has been possible. Fortunately, there has been development of mechanical processes, so that actual costs have not increased in the same ratio as intrinsic values.

Higher Steam Pressures

15 As I have said before, the production of cheap power is not dependent on the turbine alone. There is a further possible improvement in the use of higher steam pressures. The gain due to bleeding the turbine for heating feed increases with the higher pressures. With three stages of heating it is approximately from 5½ per cent at 300 lb. to 7 per cent at 1100 lb. Extra high steam

pressures render the compounding of turbines more than ever desirable because of the greater heat drop. None of us know much about the use of high pressures. We have fair knowledge of the efficiency of turbines designed to operate with moderately high pressures — up to 1200 lb. — but we know little of the mechanical behavior of the steam in the turbine. It is not certain that this high-density steam will not be quite erosive. At all events, some turbines in Europe that have operated with 450 lb. have experienced quite serious deterioration of the high-pressure portion. It happens, however, that the steam in this instance is dirty and carries considerable foreign matter. Whether the difficulty is entirely attributable to this fact is not certain. Be this as it may, it is another suggestion pointing to the desirability of high-pressure elements of low velocity for such applications. We shall know more of this shortly. There will be the Crawford Avenue Station with 550 lb. pressure. In the Weymouth Station of the Boston Edison Company an admirable experiment is being carried out of installing a 1200-lb. boiler and turbine exhausting into what is normally the high-pressure steam line in the power plant.

16 I think the desirability of a plant permanently arranged in this latter manner has been suggested where extra high-pressure turbines and boilers in combination with certain normal-pressure turbines would carry the base load and additional normal-pressure turbines and boilers placed in service on peaks. Such a scheme would seem to have much to commend it from the standpoint of economics.

17 Hitherto reheating has always been considered in connection with the operation of turbines designed for extra high steam pressure. Some engineers, I believe, have felt that reheaters are a necessary concomitant of extra high steam pressures because of the injurious effect of the high moisture content of the steam at the lower ranges of expansion. I never felt that this is true if stage feedwater heating means are well carried out and kept in service and the turbine arranged so that the maximum degree of water condensed during the expansion may be drawn to the heaters. The use of reheaters involves considerable complexity, not only because of taking steam from the turbine system to a place where it may be reheated and brought back again, but because of complication to the regulating mechanism of the turbine. In this connection it must be noted that there is little thermal advantage due to this reheating if the reheaters cannot be arranged for a small pressure drop, so I feel that when going to higher pressures, serious attempts will be made to eliminate the reheater for the reasons just stated and the further one of cost.

18 So far as concerns the economic gain due to the use of high steam pressure, you will find many opinions. I showed in my World Power Conference paper that it is to be expected that the efficiency of the turbine system such as was described, operating

with 700 deg. total temperature and without reheating, will be from 80.8 per cent efficiency ratio for 300 lb. pressure to about 79.4 for 1200 lb. This would mean a saving of steam consumption of 12 per cent, and therefore about 12.8 per cent less B.t.u. in the steam supplied, which does not take cognizance of reheating or stage feedwater heating — which latter, as I said, is more valuable as higher pressures are reached.

19 Sir Charles Parsons in his paper, in the case of turbines designed for 750 deg. fahr. total temperature and 29 in. vacuum, predicts that the gain over 250 lb. pressure will be:

5.45 per cent for 500 lb.	13.82 per cent for 1500 lb.
10.08 per cent for 1000 lb.	15.9 per cent for 2000 lb.

In this instance he has included reheating and stage feedwater heating, as well as allowance for slightly lower boiler-plant efficiency at the higher pressures.

20 Mr. V. Nordström, of Sweden, predicts in the case of 750 deg. fahr. total temperature, and 29 in. vacuum that the fuel saving over 285 lb. pressure will be:

10 per cent for 710 lb.	11.5 per cent for 1420 lb.

21 So far as the attitude of turbine designers is concerned, I believe none of them either here or abroad will hold back in producing steam-turbine combinations capable of operating at any of the high pressures thus far contemplated, and furthermore will produce them with ultimate success. I would say that generally, in Europe, all engineers are subscribing to extra high pressures, but I think they regard 750 deg. fahr. total temperature as high enough for materials at present commercially available.

22 There are many points regarding details of design applicable to all types of turbines whereby improvements may be made, which time does not permit me to discuss. I hope that, because of increasing fuel costs, we in this country will strive for higher efficiencies, keeping before us the words of the doctrine I have talked about, but with RELIABILITY in larger type than Efficiency.

DISCUSSION

A. B. CLARK.[1] The author mentioned the fact that reheating was tried some time ago. About 1900, reheating was to have been tried on a machine of about 8000 kw. capacity, but before it was actually installed it was decided to add some impulse blades to the machine with the idea of getting as much benefit as reheating would have given. Unfortunately, the impulse element did not work out as expected and the whole thing was a failure. The

[1] Engineer, Sargent & Lundy, Chicago, Ill. Mem. A.S.M.E.

reheater was not put in. Had the machine been installed as originally planned, something might have come of it.

H. C. HEATON.[1] End tightening has been employed on an Allis-Chalmers machine at Waukegan, Ill. Several experiences with severe vibration showed not the slightest trouble due to radial contact of the rotating parts. The vibrations were of such magnitude that without end tightening they would have been disastrous to an ordinary type of reaction machine.

[1] Consulting Mechanical Engineer, Sargent & Lundy, Chicago, Ill. Mem. A.S.M.E.

No. 1994

TEST OF PULVERIZED-FUEL-FIRED BOILERS AT THE LAKE SHORE STATION, CLEVELAND

By John Wolff,[1] Cleveland, Ohio

Member of the Society

THE pulverized-coal-burning equipment of the Lake Shore Station of the Cleveland Electric Illuminating Company was described by Mr. W. H. Aldrich in a paper[2] presented at the 1924 Spring Meeting of the Society. Since that time complete tests have been run, the objects of which were to compare the results with guarantees and to determine the general economy and detailed characteristics of the installation.

2 Fig. 1 shows a cross-section of the boiler and furnace, and Fig. 3 a cross-section through the boiler room and preparation plant. A summary of the equipment is given below.

Boilers, Duplex Stirling..... 30,600 sq. ft.
Superheaters, B. & W..... 4,520 sq. ft.
Water screen ..... 835 sq. ft.
Economizers, Power Specialty Company..... 22,080 sq. ft.
Furnace volume, above screen..... 26,000 cu. ft.
Driers: None used. Provision is made for waste-heat driers if necessary
Mills: Raymond, 6-roll, 6 tons per hour
Secondary Air Dampers: Bailey Meter Co., 24 per unit
Burners: Lopulco, 16 per unit
Feeders: Lopulco, 16 per unit
Feeder Drive: Individual d.c. motors, 1.5 hp. each
Blowers: Primary air, No. 4½ Sturtevant
Conveyors: Screw type, 14 in. diam.
Induced-Draft Fans: No. 10 Sturtevant Cindervane

[1] Mechanical Engineer, Cleveland Electric Illuminating Company.

[2] Pulverized Fuel at the Cleveland Illuminating Company's Lake Shore Plant. Published in *Mechanical Engineering*, September, 1924, p. 519.

Presented at a joint meeting of the Cleveland Engineering Society and the Cleveland Sections of the American Society of Heating and Ventilating Engineers and The American Society of Mechanical Engineers, Cleveland, Ohio, January 13, 1925. Contributed by the Cleveland Section of the A.S.M.E.

Motors (G. E. Co).:

Voltage	Current	Hp.	R. p. m.	To drive
220	d.c.	1.5	500–1500	feeders
2200	a.c.	100	1200	blowers
440	a.c.	15	900	conveyors
2200	a.c.	100	450	mills
2200	a.c.	50	1200	mill exhausters
2200	a.c.	150	120–360	induced draft

3 The feature of this installation by which changes in coal are automatically compensated for is the control system. This is best

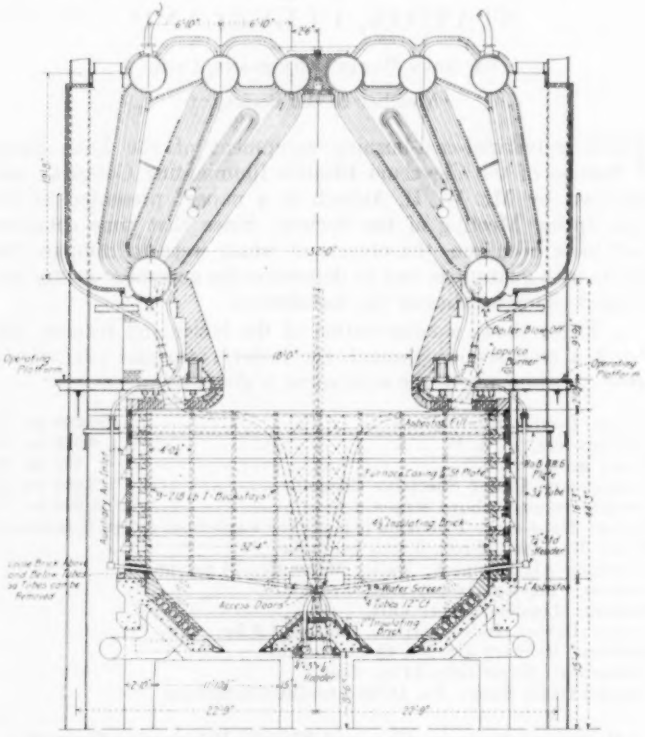

FIG. 1 CROSS-SECTION OF BOILER AND FURNACE EQUIPPED FOR BURNING PULVERIZED COAL, LAKE SHORE STATION, CLEVELAND

described as consisting of two separate controls: first, an individual control for each boiler by means of which the excess air for combustion is automatically maintained at a predetermined amount according to the boiler rating; and second, a master control by means of which the output of all the boilers is governed according to the steam pressure.

4 The individual boiler control is based on the total air flowing through each boiler, and each boiler control is now set to maintain the CO_2 as shown in Fig. 4. By this means the furnace temperature is kept approximately constant, irrespective of rating. In the actual operation of this control the induced-draft fans are

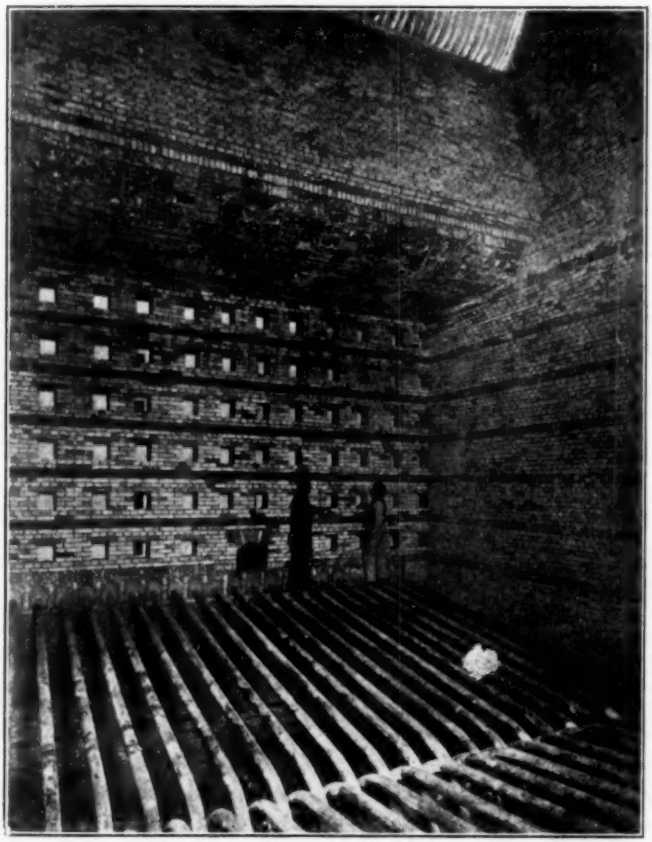

FIG. 2 INTERIOR VIEW OF FURNACE

speeded up or slowed down according to whether the excess air is too low or too high for that rating. If the rating is below the point at which the fans are used, the adjustment is made by means of a damper on the fan uptake.

5 The master control is operated by the station steam pressure. When the pressure drops below standard an impulse is given to all the boilers which increases both the coal feed and air supply,

TABLE 1 DATA OF PULVERIZED-FUEL BURNING

Test No.	13A	13	4	14	5	15	6	16
Duration, hours	22.92	21.60	26.45	23.68	18.75	25.08	24.53	21.78
Rate of driving, per cent of rating.	51	54	94	103	143	152	182	204
Coal as Fired								
Through 50 mesh, per cent	98.6	98.8	95.9	98.4	96.4	98.9	98.2	99.2
Through 100 mesh, per cent	91.1	92.5	88.3	91.3	86.5	91.2	90.4	92.6
Through 150 mesh, per cent	81.5	81.4	79.9	82.5	82.9	81.9	84.0	83.8
Through 200 mesh, per cent	72.9	69.9	75.6	73.6	79.2	72.5	79.5	74.9
Moisture, per cent	2.6	2.5	2.7	2.1	3.0	2.3	3.4	2.0
Ash, per cent	10.7	10.7	12.2	11.1	11.1	10.5	11.7	10.7
B.t.u. per lb.	12,590	12,609	12,400	12,578	12,629	12,686	12,491	12,781
Fired per hr., lb.	5,244	5,351	9,317	10,360	14,092	14,951	18,479	20,431
Per cu. ft. combustion space per hr., lb.	0.20	0.21	0.36	0.40	0.54	0.58	0.71	0.79
Water, Lb.								
Evaporated per hr.	47,582	50,584	87,956	94,523	132,257	139,847	167,959	187,540
Evaporated per lb. of coal	9.07	9.45	9.44	9.12	9.39	9.35	9.09	9.18
B.t.u. per lb. of steam	1299	1300	1307	1332	1342	1356	1357	1366
Temperatures, Deg. Fahr.								
Air to furnace	99	96	85	91	77	85	70	82
Air to feeders	97	90	91	99	91	105	89	104
Gases leaving boiler	409	425	457	467	519	530	564	587
Gases leaving economizer	169	174	203	203	230	235	252	263
Water to economizer	125	123	143	137	147	151	143	152
Water to boiler	205	215	219	218	238	242	247	255
Superheated steam	571	573	587	638	656	686	687	706
Pressure, Lb. per Sq. In. Abs.								
Boiler	253	253	251	255	254	257	256	262
Superheater	251	251	250	252	252	254	254	256
Drafts, Inches of Water								
Furnace	0.18	0.22	0.23	0.29	0.25	0.35	0.36	0.47
Leaving boiler	0.06	0.02	0.06	0.10	0.27	0.33	0.55	0.78
Leaving economizer	0.06	0.17	0.57	0.34	0.99	1.16	1.98	2.56
Air to feeders, pressure	6.4	6.1	8.8	6.9	8.2	8.4	9.8	10.2
Percentage Analysis of Gases								
CO_2 leaving boiler	13.0	12.0	15.2	15.3	14.6	15.0	14.5	14.7
O_2 leaving boiler	5.8	7.1	3.1	3.5	3.9	3.5	4.0	3.7
CO leaving boiler	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
CO_2 leaving economizer	13.0	12.0	15.1	15.0	14.3	14.5	14.1	13.8
O_2 leaving economizer	5.8	6.9	3.3	3.4	4.2	3.8	4.5	4.7
CO leaving economizer	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Heat Account in Per Cent								
Heat absorbed by boiler and superheater	81.1	83.7	85.3	83.1	84.4	84.5	83.0	82.0
Do., boiler, superheater and economizer	86.9	90.6	91.0	89.0	91.2	91.2	90.7	89.5
Loss in dry gases	1.8	2.1	2.6	2.5	3.5	3.5	4.2	4.4
Loss in water vapor	3.6	3.0	4.1	4.0	4.0	4.1	4.3	4.1
Loss in radiation, combustible ash, and unaccounted for	7.7	3.7	2.3	4.5	1.3	1.2	0.8	2.0
TOTAL	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0
[1] Duration of banking, hours	....	....	....	....	....	....	....	....

TESTS OF NO. 87 BOILER, MAY-JULY, 1924

				Starting Tests		Banking Tests				Bailey Control Tests	
9	7	8	19	12	18	22	22A	3	10-11	23A	23
18.67	23.93	30.27	24.98	36.28[1]	25.82[1]	26.90[1]	24.68[1]	28.05[1]	47.95[1]	28.17	20.28
214	228	242	270	87	142	105	108	133	141	122	134
97.8	98.8	97.7	98.3	98.8	98.2	98.9	98.8	97.8	98.9	98.8	98.8
91.4	91.9	90.1	91.6	92.0	90.1	90.9	90.2	90.2	90.8	90.1	90.8
84.7	85.5	83.3	82.2	82.5	81.2	81.7	80.9	85.0	80.6	80.5	81.7
77.8	77.9	75.5	74.4	72.4	72.6	72.9	72.1	79.9	70.8	71.1	73.0
1.7	2.2	2.4	2.0	2.2	1.7	2.4	2.3	2.6	1.9	2.5	1.7
12.6	13.1	12.8	10.7	11.1	10.9	10.5	10.5	11.9	11.8	10.3	10.4
12,473	12,283	12,371	12,717	12,619	12,694	12,587	12,684	12,399	12,543	12,684	12,738
21,757	23,702	25,205	27,445	9,785	15,608	10,154	10,874	13,341	14,170	12,160	13,512
0.84	0.89	0.97	1.06	0.38	0.60	0.39	0.42	0.51	0.55	0.47	0.52
198,591	210,921	225,578	254,357	81,804	135,175	97,420	100,371	126,224	132,127	113,596	124,123
9.13	8.90	8.95	9.27	8.36	8.66	9.59	9.23	9.46	9.32	9.34	9.19
1356	1366	1362	1353	1310	1333	1320	1323	1322	1328	1345	1348
80	73	85	77	88	86	101	96	73	91	84	81
93	93	99	100	96	107	109	100	93	92	100	101
596	621	631	646	469	524	468	481	502	507	517	532
259	274	281	318	204	242	213	209	218	229	231	237
144	147	146	153	141	150	140	133	157	145	144	136
255	259	265	265	221	256	221	219	242	239	243	243
686	706	698	681	593	640	613	622	615	628	664	670
256	259	260	270	254	259	255	254	253	252	256	256
254	257	257	265	252	255	254	253	251	252	255	255
0.35	0.39	0.38	0.58	0.24	0.37	0.28	0.28	0.20	0.26	0.30	0.30
0.77	0.94	1.21	1.54	0.07	0.50	0.09	0.10	0.24	0.24	0.26	0.34
2.68	3.18	4.08	5.15	0.28	1.63	0.41	0.45	0.82	0.76	0.91	1.17
9.3	10.2	10.3	9.9	6.3	8.7	8.2	9.7	7.6	8.8	8.2	8.4
14.5	14.5	14.3	14.2	14.9	12.3	15.5	15.4	15.1	14.5	13.3	13.7
4.0	3.9	4.3	4.3	3.8	6.4	3.2	3.0	3.4	3.9	5.6	5.3
0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
14.2	14.0	13.8	14.0	15.0	12.3	14.8	15.1	14.4	14.1	12.7	13.2
4.6	4.4	4.9	4.7	3.3	6.6	2.9	3.3	3.5	4.3	5.5	5.0
0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
82.9	82.5	81.6	81.5	74.3	75.6	86.2	82.8	84.6	83.3	83.4	81.9
91.1	90.6	90.3	89.8	79.6	82.9	92.3	89.0	91.2	90.4	90.8	89.7
4.2	4.8	4.7	5.8	2.6	4.2	2.5	2.5	3.4	3.2	3.9	3.9
4.0	4.1	4.2	3.6	4.0	4.0	4.0	4.0	4.2	4.0	4.2	3.9
0.7	0.5	0.8	0.8	13.8	8.9	1.2	4.5	1.2	2.4	1.1	2.5
100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0
....	....	....	....	10.28	3.55	5.17	5.12	11.47	9.60	....	....

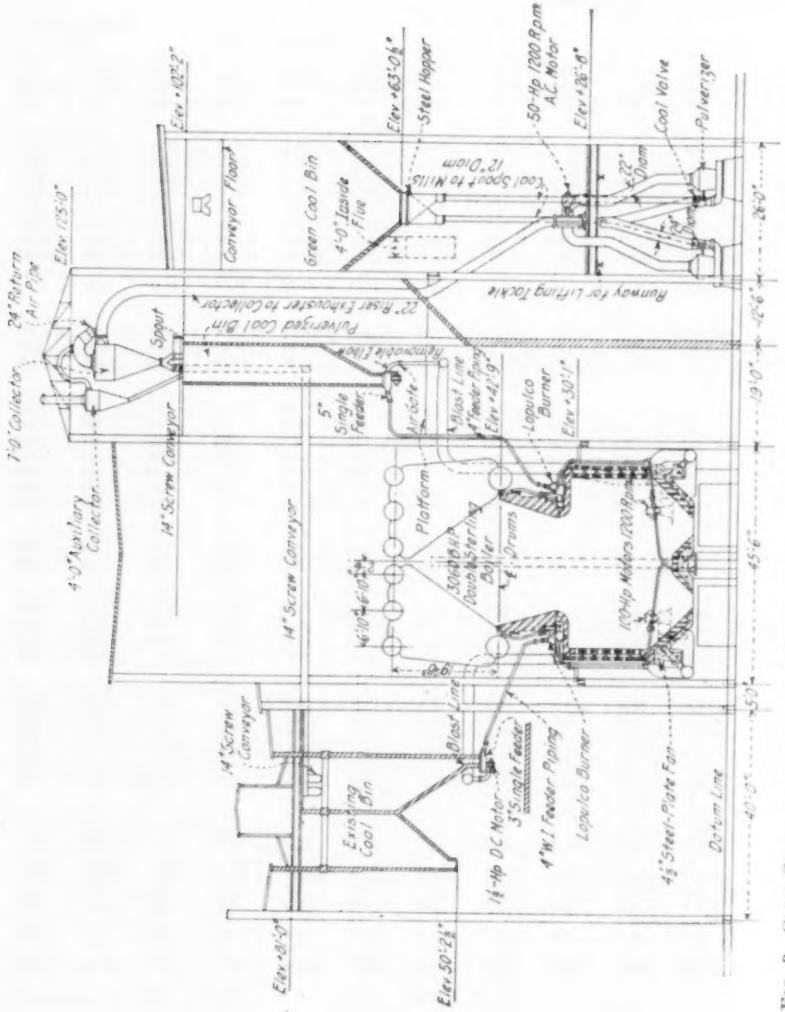

FIG. 3 CROSS-SECTION THROUGH THE BOILER ROOM AND FUEL-PREPARATION PLANT, LAKE SHORE STATION

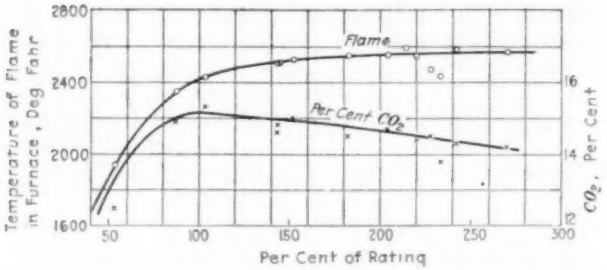

FIG. 4 FURNACE FLAME TEMPERATURES AND CO_2 CONTENTS, AT VARIOUS RATES OF DRIVING

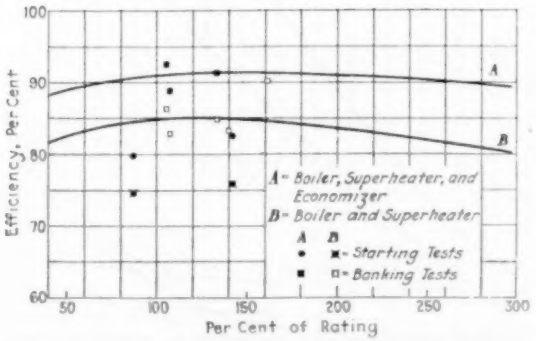

FIG. 5 EFFICIENCIES AT VARIOUS RATES OF DRIVING, CONSTANT RATING AND AUTOMATIC CONTROL

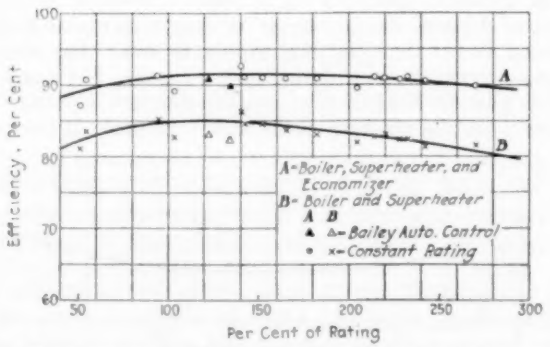

FIG. 6 EFFICIENCIES AT VARIOUS RATES OF DRIVING, STARTING AND BANKING TESTS

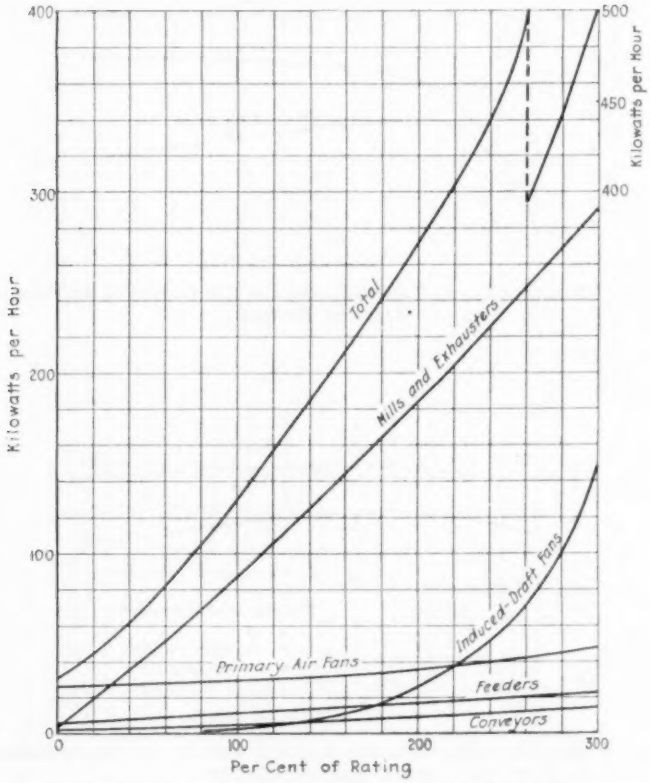

FIG. 7 PREPARATION ENERGY, KILOWATTS PER HOUR AT GENERATOR

and the opposite occurs for an increase in pressure. For changes in rate of driving, the air supply is always increased faster and decreased slower than the coal supply in order that there shall always be an excess of air during such a change. For a small drop in steam pressure the boilers receive intermittent impulses of short duration until the pressure comes back to normal. If the pressure

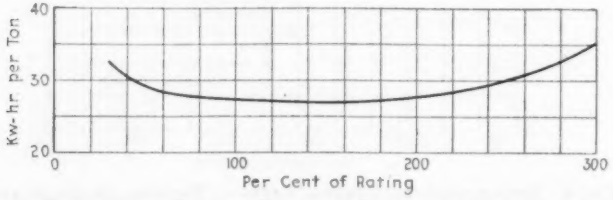

FIG. 8 PREPARATION ENERGY, KILOWATT-HOURS AT GENERATOR PER TON

drops further instead of coming back to normal, the intermittent impulses become of longer duration according to the amount of pressure drop. The duration of the intermittent impulse depends on the amount of the pressure drop.

6 This system has proved entirely satisfactory and is not subject to bad "hunting."

7 The method of conducting the tests was as follows: Both

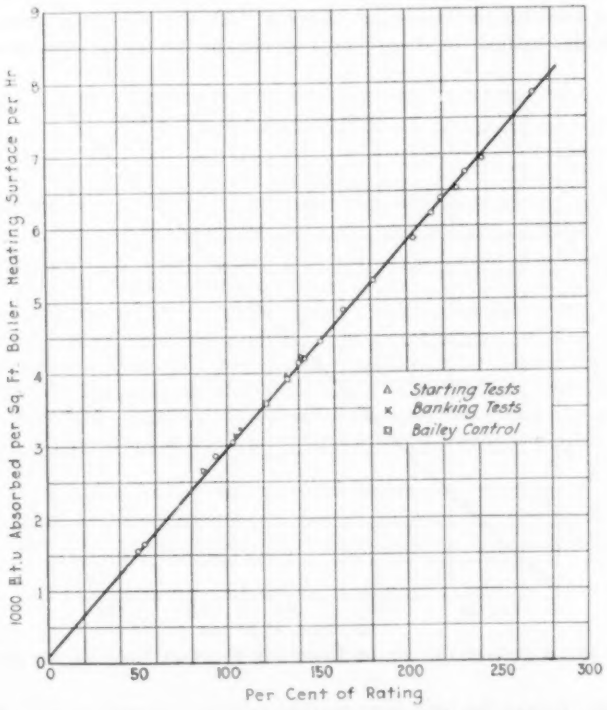

FIG. 9 B.T.U. ABSORBED PER SQ. FT. OF BOILER HEATING SURFACE PER HOUR

coal and water were weighed by balance scales which were checked by standard weights before and during the tests. Necessary precautions were taken to make certain that there was no possibility of water leakage by the installation of blanks and "telltales," and routine inspections of these were made every two hours. The coal was weighed and sampled as it entered the powdered-coal bins. Except for a few shutdowns the boiler was under continuous testing. The finish of one test was the start of the next test. Each test was approximately 24 hours long. At the conclusion of each

test the coal in the powdered-coal bins was leveled off at about three feet above the feeders.

8 All thermometers were calibrated before the tests, and the more important ones at different times during the tests.

9 Due to the great width of the boiler, CO_2 samples were taken at five points across each boiler uptake. Each point was sampled separately in rotation from a collecting bottle which had been filling while the previous sample was being analyzed. Two Orsats were used for the boiler gas samples. CO_2 samples were taken in a similar manner at three points across the uptake of each econo-

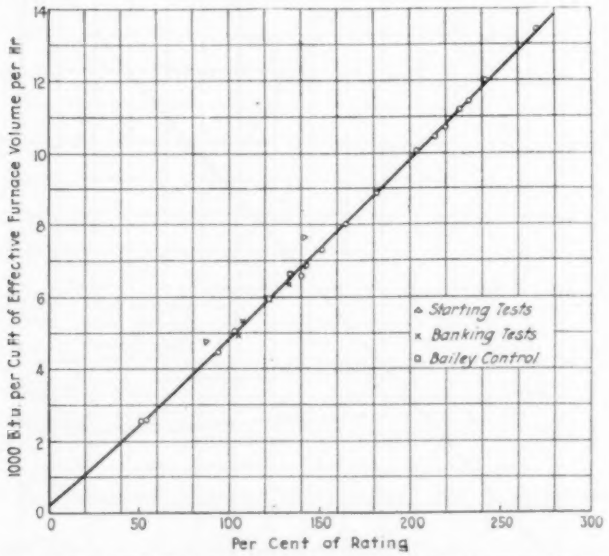

FIG. 10 B.T.U. PER CU. FT. OF EFFECTIVE FURNACE VOLUME PER HOUR

mizer. Before the tests were started velocity traverses at different ratings were taken in the boiler uptake and the economizer outlet in order that the gas samples and temperatures should represent the weighted average as near as possible.

10 With the exception of two of the tests the firing was manual. The CO_2 for each rating was carried at the highest point possible without damage to the furnace walls. This resulted in the curve shown in Fig. 5. Since the tests, improvements in the end-burner operation and air distribution have been made which make it possible to carry slightly higher CO_2 than shown in Fig. 4.

11 Two tests, using automatic control, were run with the boiler operating under plant load conditions, during which periods

the rate of driving varied from 90 to 240 per cent of rating, the resultant efficiencies conforming very closely to test results based on a weighted average for the rate of driving in question.

12 The results of these tests should also be considered as the results of actual operation since the test results are automatically maintained in every-day operation irrespective of changes in coal.

13 Plant records show the following monthly efficiencies for this installation.

	Efficiency, per cent	
	Gross	Net
July	87.8	85.9
August	90.1	88.2
September	90.4	88.3
October	88.4	86.8
November	90.0	88.2

14 The coal weights for the above are taken from track scales and the coal sample is taken just after the coal is crushed. According to recent tests the track scales show 0.50 to 1.00 per cent more coal (on a dry basis) than the pulverized-coal scales used on the test. Most of this difference is due to moisture lost in process of pulverization, but the monthly efficiencies shown above are not corrected for this amount. The net efficiency is calculated from the total energy used in coal preparation (exclusive of crushing, i.e., reduction in size from mine run to slack), conveyors, feeders, and forced- and induced-draft fans. In other words, the coal equivalent for the above operation is charged against the boiler.

15 No special coal was used for these tests, the supply being the regular plant supply from several different shippers and some market coal. A sample analysis on the as-fired basis is as follows:

Proximate Analysis	
B.t.u.	12,629
Moisture, per cent	2.3
Volatile, per cent	35.4
Fixed carbon, per cent	51.2
Ash, per cent	11.1
	100.0

Ultimate Analysis	
Moisture, per cent	2.3
Carbon, per cent	68.9
Hydrogen, per cent	4.8
Nitrogen, per cent	1.1
Oxygen, per cent	8.1
Ash, per cent	11.1
Sulphur, per cent	3.7
	100.0

16 The fusion temperature of the ash of the coal used as compared with that of the brick is shown below. In actual operation

the flame has to be kept away from the furnace wall, for the brick is eroded at temperatures above the fusion temperature of the coal ash.

	Softening point, deg. fahr.	Me ting point, deg. fahr.
Plaster coating	2598	2640
Brick	2585	2650
Coal ash	2016	2147

17 In order to determine the schedule of operation of these boilers in connection with the rest of the plant, several banking tests and two starting tests were run. Fig. 6 shows that for a 5-10-hour bank there is no appreciable loss in efficiency. To bring one boiler unit from room temperature to that corresponding to 100 or 150 per cent of rating, considerable heat is absorbed by the structure which is not used in generating steam. As calculated from the drop in efficiency of these starting tests below the curve (Fig. 6), about 26 tons of coal are lost in bringing one of these boilers from room temperature up to the temperature corresponding to 100-150 per cent of rating.

18 Temperatures and miscellaneous firing data are shown in Table 1.

19 Before the tests were started, the equipment was given a complete inspection. The economizer appeared clean, but the draft loss subsequently showed that there must have been considerable deposit on the tubes. The economizer was cleaned after the tests and Table 2 shows the draft loss after cleaning. The draft loss shown in Table 1 is therefore not representative. Detailed tests were also run before and after cleaning to determine the change in heat transfer. These tests showed that the deposit on the outside of the tubes was of such nature as to affect the draft loss more than the heat transfer.

TABLE 2 DRAFT LOSS AT VARIOUS RATINGS

Per cent of rating.....	100	125	150	175	200	225	250	275	300
Draft loss, inches of water	0.09	0.18	0.30	0.45	0.60	0.75	1.05	1.40	1.90

20 Figs. 7 and 8 show the preparation energy per ton and also for different boiler ratings.

No. 1995

THE LOCOMOTIVE TESTING PLANT AND ITS INFLUENCE ON STEAM-LOCOMOTIVE DESIGN

By Lawford H. Fry,[1] Burnham, Pa.
Member of the Society

The paper gives a brief history of locomotive testing plants and their contributions to the development of the locomotive. The work done by the Altoona testing plant is told in brief references to the results of tests as reported in various bulletins published by the Pennsylvania Railroad. Contrasts are made of size and economy of locomotives twenty years ago when the first testing plant was built and the present time. One appendix to the paper develops the method used in determining locomotive efficiencies, and another is an extensive bibliography of locomotive testing.

IT IS appropriate that this subject should be considered by this Society at Altoona. To mechanical engineers the locomotive should be interesting not only for the many problems of design, construction, and operation which it presents, but also because without the cheap transportation which the steam locomotive has furnished there would have been no opportunity for the growth of the industries by which and with which the whole profession of engineering has developed. No study of locomotive development and operation can be complete without consideration of the work done by the locomotive testing plant at Altoona, as the results of this work have materially influenced locomotive design not only in this country, but abroad. Rowland (46)[2] is designing locomotive boilers in England according to formulas developed largely from test data from the Altoona plant, and his formula for efficiency of combustion had been adopted by the Railway Board of India as a basis for their official method of estimating boiler capacity (51).

[1] Metallurgical Engineer, Standard Steel Works Company.
[2] Numbers in parentheses refer to the Bibliography, Appendix No. 2.

Presented at the Regional Meeting, Altoona, Pa., October 5 to 7, 1925, of The American Society of Mechanical Engineers.

Locomotive Testing Plants

2 In view of the complete descriptions of the various locomotive testing plants which have been presented to the Society (2) (19) or published elsewhere (20) (41), it is unnecessary to give a detailed description here. The essentials of a locomotive testing plant are (*a*) that the locomotive driving wheels be supported on carrying wheels fitted with friction brakes to absorb the energy developed, and (*b*) that a dynamometer be provided to measure the drawbar pull. In the latest plants the friction brakes are of the Alden hydraulic disk type, which has proved to give satisfactory absorption of power and furnishes sensitive and accurate control of the speed.

3 On a modern testing plant it is possible to operate any given locomotive through its whole range of power. Within this range any desired combination of conditions, such as speed, cut-off, etc., can be selected and maintained constant during a run of an hour or more, and while running under such conditions measurements can be made with laboratory accuracy and completeness. It is this combination of constancy of conditions and accuracy of measurements which gives the results obtained on a testing plant their great advantage over road tests of locomotives. Road tests with a dynamometer car are valuable for checking and completing information obtained on the plant as to drawbar pull, but in all questions of steam production and consumption the plant results are far more authoritative. There is much detailed information which now controls locomotive design which it would have been practically impossible to obtain without a testing plant.

4 The locomotive testing plant is purely of American origin, and though the results of the tests made here have been widely studied, the author has been unable to find a record of any real testing plant installed outside of the United States. Appendix No. 2 gives a fairly comprehensive bibliography of papers, articles, or other publications dealing with or deriving data from the various American plants. Locomotive-testing-plant history goes back to 1891, when the first plant was designed and erected at Purdue University under the supervision of Dr. W. F. M. Goss. A description of this plant was presented to the Society in 1892 by Dr. Goss. Locomotive testing plants have therefore been in use for more than 33 years, or one-third of the century during which steam locomotives have been employed in America. The growth in locomotives during that time may be measured by the fact that the first locomotive tested at Purdue was a 4-4-0 type, a representative passenger locomotive of that date, weighing in working order about 85,000 lb., while the most recent passenger engine tested on the Altoona plant has a weight of 309,000 lb., or more than three and one-half times as much.

5 Altogether there are records of six locomotive testing plants built and operated. The first Purdue plant, referred to above, was

destroyed by fire in 1894, but was rebuilt and enlarged the same year. While it was out of commission the late Robert Quayle of the Chicago & Northwestern Ry. installed a temporary plant at Kaukauna, Wis., to carry out tests for the A.R.M.M.A. committee on exhaust nozzles. In the following year, 1895, Mr. Quayle built a permanent plant at the Chicago & Northwestern Ry. shops in Chicago. In 1899, as a result of the gift by the Baldwin Locomotive Works of a Vauclain compound locomotive, Columbia University built a testing plant under the direction of Prof. F. R. Hutton.

6 These plants were all of small capacity, and those of the C. & N. W. Ry. and of Columbia University have been abandoned. The first modern locomotive testing plant capable of handling locomotives of various designs was that of the Pennsylvania Railroad built in 1904. This was designed by the late Axel Vogt, then chief mechanical engineer, assisted by W. F. Kiesel, Jr., the present chief mechanical engineer of the Pennsylvania System. This plant was first installed at St. Louis for the Louisiana Purchase Exposition of 1904 and after operating there throughout the Exposition was transferred to its present location in Altoona.

7 The latest plant is that at Illinois University and was built in 1914 under Dr. W. F. M. Goss as Dean of the College of Engineering and Prof. E. W. Schmidt in charge of the Department of Railway Engineering. This plant follows the general design of the plant at Altoona, but provides elaborate arrangements for catching the sparks and cinders thrown out of the stack. For this purpose the exhaust gases are drawn by an exhaust fan through a duct in the roof of the building and are discharged through a centrifugal separator to a 45-ft. stack. The cinders escaping from the locomotive stack are carried by the sweep of the gases through the duct, and those trapped in the separator can be collected and weighed and analyzed.

8 Details of design of the various plants will be found in various publications listed in the attached bibliography, the earlier plants in Dr. Goss' paper (19) presented to the Society in 1904, the Pennsylvania plant in publications of the Pennsylvania Railroad (20), and the Illinois plant in the Engineering Experiment Station Bulletin No. 82 (41).

Work Done by Various Plants

9 The work done on these plants has been of varying importance. The Chicago & Northwestern Ry. plant was of considerable service to the early committees of the A.R.M.M.A. in work on exhaust nozzles, but published no other work. The Columbia University plant, situated at Morningside Heights without railroad connections, was adapted only to use the university locomotive for student-instruction purposes. The Illinois University plant has

published two reports on tests, one (41) describing complete tests on a single-expansion consolidation locomotive, and the other (45) covering boiler tests on six different sizes of Illinois coal.

Work of Purdue Plant

10 The widest range of work has been done on the Purdue and the Pennsylvania plants. This can be seen by reference to the bibliography in Appendix No. 2. At Purdue some work of the A.R.M.M.A. committee on locomotive front ends (23) was carried out between November, 1905, and April, 1906, with an Atlantic-type locomotive belonging to the New York Central R. R. Practically all of the other work has been done with the two experimental locomotives of Purdue University. Both were of the 4-4-0 type. The first, purchased in 1891 as a representative passenger locomotive of that date, weighed about 85,000 lb. in working order. It was replaced in 1897 by a more modern engine weighing about 109,000 lb. with about 61,000 lb. on drivers. Ten years later, in 1907, a Cole superheater was applied to this locomotive; in 1909 the superheating surface was reduced, and in 1911 the locomotive was overhauled and a Schmidt superheater applied (52). Very many phases of locomotive operation were studied with these locomotives, and the data secured were given freely in papers presented to the Society and other technical bodies as shown in the bibliography.

11 Writing in 1904 (19) of the information contributed by the Purdue plant, Dr. Goss pointed out that the plant had thrown light on the factors limiting indicated horsepower and rates of combustion which were previously imperfectly understood; that it had shown how the relation between cut-off and steam consumption varied with the speed; and that it had been used to study the action of the exhaust in producing draft and had thus assisted in determining the best proportions for locomotive front ends.

12 The early work done at the plant consisted in operating the 4-4-0 type locomotive under varying conditions of speed and load. The information obtained was general in character and bore chiefly on variations in tractive force and steam consumption as affected by changes in speed and cut-off. The effect of rate of operation on boiler efficiency was also studied, but as smokebox-gas analyses were not made, no great progress resulted in this direction. A considerable body of definite information was made available, which was undoubtedly of general value to locomotive designers in determining correct sizes for locomotives to perform a given service, and in securing a proper balance between cylinder power and boiler capacity, but such information was merely supplemental to much of a similar nature that had been secured by road tests, and no basically new information was brought to light. The front-end and stack tests, however, covered research work

which could not have been carried out except at a locomotive testing plant. Certain principles previously unknown were brought to light and American locomotive design was definitely affected. It was shown that a tapered stack was preferable to a straight one, and that while a draft pipe would improve the action of a stack which was too small, no combination of draft pipe and stack would give better results than a properly proportioned stack without a draft pipe. It was also shown that the draft-producing action of the exhaust steam was independent of the intermittency of the exhaust, the essential factor in the suction produced being the quantity of steam exhausted. This last fact has been confirmed by all later test-plant results but is often overlooked, particularly when the claim is made that a multi-cylinder locomotive gives a more efficient exhaust action. The results of the various front-end tests were embodied in a series of equations indicating the best proportions for stack and front ends. These equations, while largely empirical, provided a logical basis for the design of the front end and had a considerable influence on American locomotive practice.

13 The last published information from the Purdue plant dates back fifteen years or more to two reports (24) (27) published respectively in 1907 and 1910 by the Carnegie Foundation. These reports cover two series of tests made by Dr. Goss with the Purdue locomotive, studying, in the earlier series, variations in boiler pressure, and in the later, variations in superheat. The effect of such variations on boiler and engine operation is shown. In the superheat series (27) fairly accurate heat balances are given, which confirm the information obtained earlier from the Pennsylvania Railroad plant. The two series represent general research work and give definite information on certain points which is of value to the student of locomotive operation.

Work of the Pennsylvania Railroad Test Plant

14 The installation of the Pennsylvania Railroad locomotive testing plant at the St. Louis Exposition in 1904 opened a new and important era in locomotive testing. The earlier plants had been adapted only for testing light four-coupled locomotives. The Pennsylvania plant was capable of handling locomotives of much greater weight and more varied design. Immediate use was made of the greater plant capacity. The series of tests made at St. Louis covered eight locomotives of widely different designs, and the data secured and published (20), Locomotive Tests and Exhibits, 1905, were far more complete than any previously available. This test furnished for the first time sufficient information to enable heat balances to be drawn up for a locomotive boiler. Such balances were published by the author in 1908 (29), and gave for the first time exact knowledge as to the relative importance of the various losses which determine the efficiency of the locomotive boiler. Since

its transfer to the present location in Altoona the plant has done much work in testing new locomotive designs as produced, and in providing a constantly accumulating mass of information which has made possible continued improvements in design. The greater part of the activity of the Altoona plant has been carried out under J. T. Wallis, who was appointed general superintendent of motive power in January, 1912, and who since March, 1920, has been chief of motive power. The direct charge of the test plant was in the hands of C. D. Young, engineer of tests, from November, 1911 to May, 1917, and is now in the hands of his successor F. M. Waring.

Results Obtained on the Altoona Test Plant

15 The very great influence which the work done on the Altoona plant has had on locomotive design is indicated by the following statements as to work done and conclusions reached. These are abstracted from the various Pennsylvania Railroad test-plant bulletins as shown by the reference numbers.

16 *Bulletin No. 9.* A self-cleaning front end was developed with diaphragm, exhaust nozzle, and stack arrangement, which gave better results than the form recommended by the Master Mechanics Association.

17 *Bulletin No. 15.* After test of an underfeed stoker it was recommended that for each 1900 lb. per hr. of water evaporated there be provided at least one square foot of free air space through the fuel-bed support.

18 *Bulletin No. 21.* A comparison was made between results obtained with long and short boiler tubes, the boilers compared being those of the classes K-2sa and E-6s, having tube dimensions respectively as follows: Length, 21 ft. 0 in. and 13 ft. 8 in.; outside diameter, $2\frac{1}{4}$ in. and 2 in.; ratio of length to inside diameter, 125 and 94. When operating with a smokebox vacuum of 6 in. of water the boilers when compared on the basis of evaporation per square foot of heating surface and efficiency gave the following figures:

Long tubes, evaporation 10.8 lb. per sq. ft.; boiler efficiency 64 per cent
Short tubes, evaporation 14.7 lb. per sq. ft.; boiler efficiency 57 per cent

A short tube gives free steaming or rapidity of evaporation at the expense of efficiency, so that the most desirable tube length is found by practical considerations as a result of balancing steam production against boiler efficiency. On this basis and taking into account details of construction the Pennsylvania Railroad adopted as standard for 2-in. tubes a length of 15 ft. or 102 inside diameters, and for $2\frac{1}{4}$-in. tubes a length of 19 ft. or 113 inside diameters. The conclusions arrived at were checked against and found to be substantially in accord with those derived from the early experiments made by M. A. Henry on the Paris, Lyons & Mediterranean Ry.

with much lower rates of evaporation. It was also established that the rate of evaporation is determined by the weight of gases passed over the heating surface and is limited by conditions of combustion and not by a failure of the heating surface to absorb heat.

19 *Bulletin No. 28.* The maximum rate of evaporation was established as 7000 lb. of water per hour per square foot of fire area of the tubes.

20 *Bulletin No. 23.* It was shown that for piston valves a diameter of 12 in. was sufficient for cylinders up to 27 in. in diameter when using superheated steam, and subsequent tests showed that valves of this diameter could be used with 30-in. cylinders with cut-off limited to half-stroke. Measurements of the stresses in the valve stems gave definite figures showing the great advantage in lightened valve gear to be secured by using valves of the minimum size.

21 *Bulletin No. 24.* An extensive series of experiments on superheaters of various designs produced data which are summed up in Fig. 1, showing engine steam consumption for varying conditions of superheat, cut-off, and speed. This diagram gave for the first time, unobscured by other variables, authentic information as to the effect of superheat on steam consumption.

22 *Bulletin No. 27.* As a result of earlier tests on the plant certain changes in design of the E-6s type of locomotive were recommended, as follows:

a An exhaust nozzle giving better draft conditions than the plain circular form
b A screw type of reversing gear
c An enlargement of the cylinders to such a size that the maximum horsepower may be developed at a cut-off not exceeding 30 per cent
d An increase in tube length
e The application of 12-in.-diameter piston valves
f A reduction in the width of the center grate bearer.

Tests of the locomotive redesigned as recommended gave a maximum evaporation 15 per cent greater, combined with a boiler efficiency 9 per cent better, than that of the original engine. At the same time the engine performance was generally better, so that a higher drawbar pull was developed at all speeds and the maximum drawbar horsepower was increased over 20 per cent. These advantages were secured with an increase in total weight of only 2.5 per cent. Careful design of the reciprocating parts enabled them to be kept down in weight so that the dynamic augment at 70 m.p.h. was less than 30 per cent of the static load on the drivers.

23 *Bulletin No. 29.* Tests were made with a K-4s Pacific-type locomotive built in 1914 in accordance with recommendations and experience obtained with locomotives of earlier designs on the testing plant. The coal and water rates and the high thermal

efficiency showed this to be the most economical passenger locomotive so far tested on the test plant. For any given amount of fuel fired this locomotive developed more power than any engine previously tested.

24 *Bulletin No. 30.* Accurate tests demonstrated the advantage gained in boiler efficiency by the use of a firebrick arch when high-volatile coal is being burned.

25 *Bulletins Nos. 31 and 32.* The testing plant having shown definitely the great difference in steam consumption between cut-off at full stroke and cut-off at half-stroke, it was logical to attempt to produce a freight locomotive which, even in low-speed drag

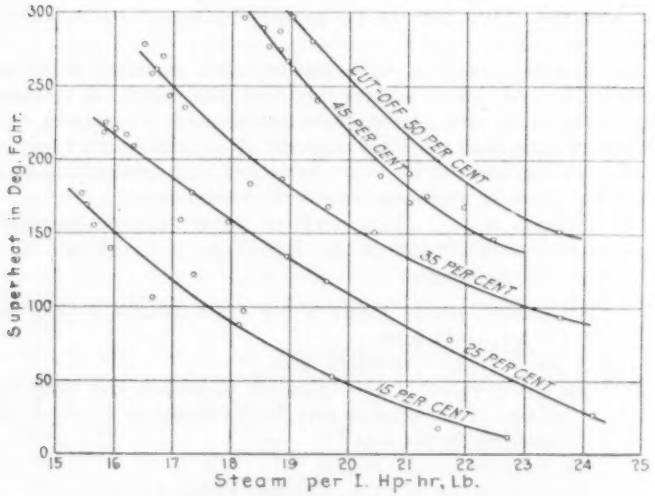

FIG. 1 EFFECT OF SUPERHEAT ON STEAM CONSUMPTION AT VARIOUS SPEEDS AND CUT-OFFS

service, could operate at cut-offs of not more than 50 per cent. The result was the development of the I-1s class locomotives with a boiler pressure of 250 lb. per sq. in. instead of 205 and with cylinders enlarged to enable full power to be developed without using a cut-off longer than 50 per cent of the stroke. Compared with the previous standard freight locomotive the new engine with an increase of 16 per cent in weight gave an increase of 25 per cent in power, and in full gear at low speed showed a reduction of 38 per cent in steam used per indicated horsepower. This type of locomotive with feedwater heater added, giving a further increase in 10 per cent in indicated horsepower, is now the standard freight locomotive of the Pennsylvania System.

Notes on Results Quoted in Pars. 16-25

26 The various statements in Pars. 16 to 25 which have been quoted in abstract from the P. R. R. test-plant bulletins show how knowledge has accumulated and how, as a consequence, locomotive designs have been developed and improved. In the majority of cases the tests have been directed to the study of certain definite locomotives, and have been analyzed to determine the possibility of improvements in the design of the engines tested. In a few cases, however, a number of tests have been correlated to permit general conclusions being drawn. Instances of this are the statements as to tube length in Par. 18, and as to fire area of tubes in Par 19. The latter statement to the effect that the maximum rate of evaporation was established as 7000 lb. of water per hour per square foot of fire area of tubes may require some explanation for those who have not studied the test-plant results in detail, and probably would require to be slightly qualified in the light of later test-plant experiments. Flue fire area is given as the limiting factor in evaporative capacity because the tests have shown that evaporation is primarily dependent on the amount of air (gases of combustion) taken through the boiler and is only secondarily dependent on the area of the heating surface with which the gases of combustion come in contact. If the size and style of flue are unchanged, the relations between fire area, heating surface, evaporation, and boiler efficiency may be summed up as follows:

a An increase in heating surface without an increase in fire area means an increase in tube length. This will give a slightly greater boiler efficiency, but the evaporation will be increased only very slightly, if at all, as owing to the greater resistance which the longer tubes offer to the gases it will not be possible to increase the amount of gas taken through the boiler

b An increase in fire area without an increase in heating surface means an increase in the number of tubes with a proportionate reduction in tube length. Any considerable change in this direction will give a considerable increase in evaporative capacity, because the increased fire area and reduced resistance in length will enable a considerably greater weight of air to pass through under the same draft. At the same time the shortening of the tubes will entail a slight reduction in efficiency

c If a boiler working below its maximum capacity is considered, the fire area may prove to be less important than is indicated above. Take, for example, a boiler fired at the rate of 80 lb. of coal per sq. ft. of grate while the maximum capacity is not reached with less than 120 lb. per sq. ft. of grate area. With an efficient front end it

would probably be possible to block up 20 per cent of the flues and still to draw the same amount of air through the boiler. The fire area would be reduced 20 per cent, but combustion would be at the same rate and the efficiency of heat absorption in the flues would be affected only to the extent of 1 or 2 per cent. The number of tubes being reduced, the amount of gas passing through each tube

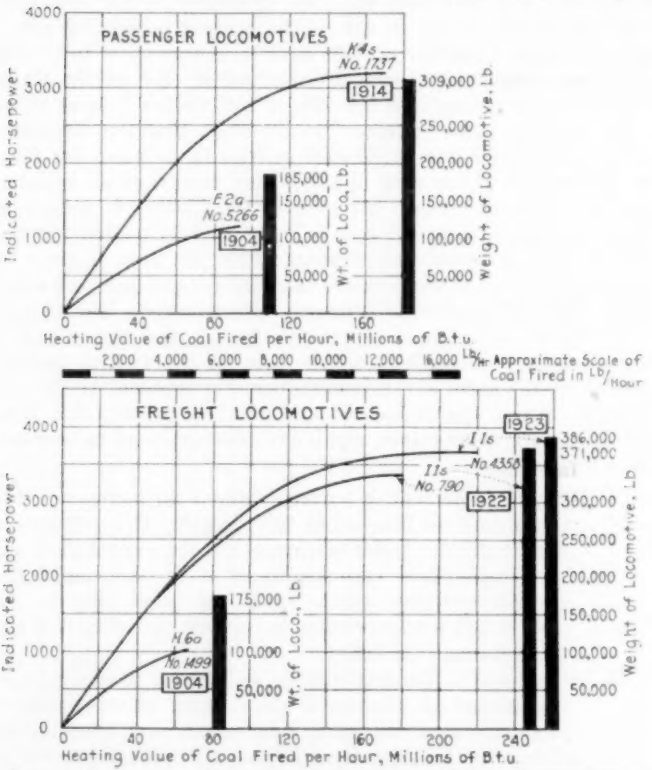

FIG. 2 GROWTH IN SIZE AND EFFICIENCY OF STEAM LOCOMOTIVES

is increased. The increase in rate of flow increases the rate of heat transfer so that the heat taken up by each flue is increased, and the total amount taken up is practically the same in spite of the reduction in the number of flues acting.

GROWTH IN LOCOMOTIVE SIZE AND EFFICIENCY

27 To attempt to give a general idea of the growth and development of the P. R. R. locomotives during the time that the

locomotive testing plant has been in operation, Fig. 2 has been prepared. It covers both freight and passenger locomotives and compares the engines which were standard in 1904 with those now standard. The heavy vertical lines are proportional in each case to the weight of locomotive in working order without tender, while the curves give the indicated horsepower in relation to the heating value of the coal fired per hour. The heating value of the coal rather than the weight has been used so that all curves could be plotted on the same base line in spite of slight differences of quality in the coal used. To aid, however, in a general consideration an approximate scale of rate of firing in pounds of coal per hour has been added. This is based on a heating value of 14,000 B.t.u. per lb., which is not far from the average value of the coals used. On this basis the figures given in Table 1 for the maximum rate of operation have been obtained.

TABLE 1 GROWTH IN LOCOMOTIVE SIZE AND EFFICIENCY

(Based on maximum rate of operation)

Service	Type	Class	Year built	Loco. weight, lb.	I. hp., max.	Coal fired, lb.	I. hp. per 1000 lb. of loco. wt.	Coal per i. hp. per hr., lb.
Passenger	4-4-2	E-2a	1904	185,000	1200	6,000	6.5	5.0
Passenger	4-6-2	K-4s	1914	309,000	3300	10,000	10.7	3.0
Increase in per cent.....							65	
Freight	2-8-0	H-6a	1904	175,000	1000	4,500	5.7	4.5
Freight	2-10-0	I-1s	1923	386,000	3530	12,800	9.1	3.6
Increase in per cent.....							60	

Attention is called to the horsepower developed per thousand pounds of locomotive weight, which, it will be seen, has been increased by 65 per cent for the passenger and 60 per cent for the freight locomotives.

28 It is evident that since the Pennsylvania Railroad installed the locomotive testing plant the locomotives have increased not merely in size but also in the efficiency with which the weight and fuel are used. Some of the steps by which the improvement in efficiency has been obtained have been noted in the abstracts from the various test-plant bulletins.

Locomotive-Boiler Efficiency

29 So far the work of the test plant has been described by abstracts from the bulletins which deal mainly with the solution of concrete problems arising from the testing of certain definite designs of locomotives. The test plant has, however, been of great value in providing material for a study of locomotive operation in general terms, and in thus advancing our knowledge of the subject. Probably the greatest advance has been made in knowledge pertaining to factors which determine boiler efficiency. The

state of knowledge before the Pennsylvania plant results had been thoroughly studied is illustrated by an article (22) by the author in 1904 and H. H. Vaughan's paper (25) before the Canadian Railway Club in 1907. Speculation was active as to the best proportions to be assigned to firebox and heating surface, but very little was definitely known. Dr. Goss's work had indicated that the losses by incomplete combustion were large, but with no

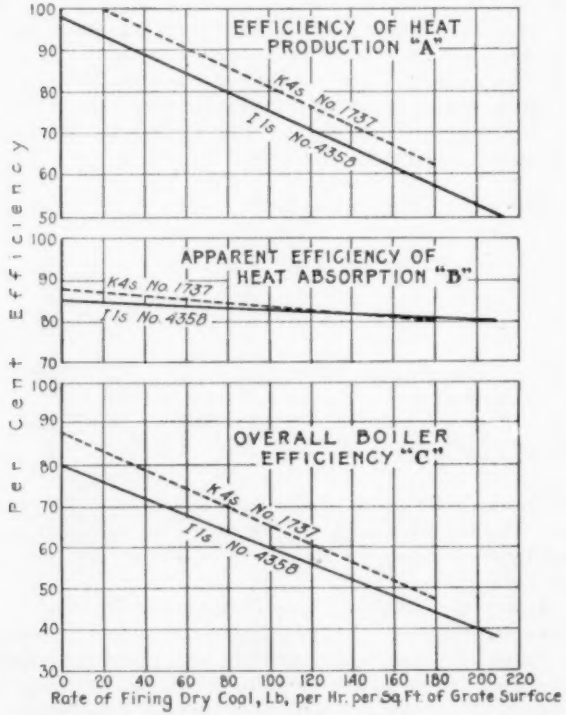

FIG. 3 LOCOMOTIVE-BOILER EFFICIENCIES IN RELATION TO RATE OF FIRING

information as to the amount of air consumed, accurate information as to the details of combustion and heat absorption was lacking. Against this set the fact that during the recent tests of the Missouri Pacific three-cylinder locomotive (54) the author was shown a predicted curve of smokebox temperatures which in the majority of cases did not show differences of more than 15 deg. fahr. from the results actually obtained.

30 As a concrete illustration of the information which can be derived from the test-plant data by further analysis, there is

given below a brief account of some work which the author has undertaken recently in connection with a study of locomotive-boiler efficiency.

31 In all tests of locomotive boilers in which the boiler efficiency is measured over a considerable range of rates of firing it has been found that as the rate of firing increases there is a drop in boiler efficiency which can be represented best by a straight-line law (47) (50). In Fig. 3-C the lines found for the Pennsylvania Railroad K-4s passenger and the I-1s freight locomotive are shown. The boiler efficiency here is the heat taken up by the boiler in evaporation and superheating expressed as a percentage of the heating value of the coal fired. The value found for this efficiency at any given rate of firing is dependent on three factors:

1 The efficiency of combustion, which is the efficiency with which the coal is burned in the firebox. In other words, the amount of heat actually produced in the firebox expressed as a percentage of the heating value of the coal fired

2 The efficiency with which the heat produced is taken up by the boiler

3 The amount of heat taken up by the boiler which is lost by external radiation. In the absence of exact information the author assumes this external loss to be 5 per cent of the heat utilized in evaporation. This assumption has been justified by the results obtained.

32 Now from the data secured at the locomotive testing plant it is possible to separate the overall boiler efficiency into its component factors. This being done for the boiler efficiencies shown in Fig. 3-C, the lines shown in Figs. 3-A and 3-B represent respectively the efficiencies of combustion and heat absorption as defined above.[1] Numerical values for the three efficiencies at representative rates of firing are given in Table 2.

TABLE 2 LOCOMOTIVE BOILER EFFICIENCIES

Locomotive	Rate of firing, lb. per sq. ft. grate per hr.	Efficiency of heat production, per cent	Efficiency of heat absorption, per cent	Overall boiler efficiency, per cent
K-4s	80	86.0	85.5	70.0
	120	76.5	83.5	61.0
	160	67.0	81.5	52.0
I-1s	80	79.7	84.4	64.0
	120	70.6	83.2	56.0
	160	61.5	82.0	48.0

33 These figures are not offered here for the purpose of making a detailed comparison of the K-4s and the I-1s locomotives, but are given as representatives of the results obtained in good locomotive practice. It is characteristic of the locomotive boiler to show efficiency relations generally similar to those in the table. The

[1] See Appendix No. 1.

efficiency of heat absorption shows high values which remain nearly constant, being little affected by the rate of firing. The efficiency of combustion shows a large variation with the rate of firing. At low rates the efficiency is very close to 100 per cent, but falls rapidly as the rate of firing is increased. At a normal maximum rate of from 100 to 120 lb. per square foot of grate per hour the efficiency of combustion is in the neighborhood of 60 per cent, while if the boilers are pushed to actual maximum power the efficiency falls to less than 50 per cent. The efficiency of heat absorption is, as has been said, much more uniform, never dropping below 80 per cent at the highest rate of firing and showing values of about 86 per cent at low rates of firing.

34 Now it is evident that if the overall efficiency of the boiler is to be improved it is desirable to study closely the two component efficiencies. In the first place, it is noted that at normal rates of firing the efficiency of combustion is in the neighborhood of 60 per cent, and the efficiency of heat absorption is over 80 per cent. This indicates that there is a greater margin for improvement in combustion than in absorption. In reality, the possible margin for improvement in absorption is even less than would appear from these figures, for while it is possible to have 100 per cent efficiency of combustion with low rates of firing, it is not physically possible for the efficiency of heat absorption, as defined above, to have a higher value than about 91 per cent. This follows because it is physically impossible, even under ideal conditions, for the boiler to take up all of the heat produced in the firebox. Under no conditions is it possible to cool the smokebox gases below the temperature of the saturated steam in the boiler. With a boiler pressure of 205 lb. per sq. in. this gives an ideal minimum temperature of 390 deg. fahr for the smokebox gases, at which temperature each pound of gas will carry 96 B.t.u. Under normal conditions the production of one pound of combustion gases will be accompanied by the liberation of approximately 1080 B.t.u., and with an atmospheric temperature of 65 deg. fahr. adding a further 13 B.t.u., the total heat offered to the heating surface will be approximately 1093 B.t.u. per pound of gases of combustion. With ideal conditions by which the gases of combustion were cooled to the saturated-steam temperature, 96 B.t.u. would be left in, and 997 B.t.u. absorbed from, each pound of gas. That is, under ideal conditions the heating surface cannot possibly take up more than 91 per cent of the original heat in the gases. Let us designate as "apparent efficiency" the efficiency of heat absorption based on heat produced, and designate as "real efficiency of absorption" the heat taken up expressed as a percentage of the heat absorbable under ideal conditions. Then while the figures above show "apparent" efficiencies of heat absorption ranging from 80 to 86 per cent, the "real" efficiencies range from 94 per cent at low to 88 per cent at high rates of firing. It is evident that

as a piece of heat-interchange apparatus the modern locomotive boiler, as exemplified by the Pennsylvania Railroad boilers in question, is highly efficient and that there is but a small margin for improvement in heat absorption.

35 The efficiency of combustion shows a different condition. Here an efficiency of 100 per cent is possible and is shown by the tests to be reached at low rates of combustion. At the usual rates of operation, however, the values are generally in the neighborhood of 60 per cent or less.

36 There is evidently considerable field for improvement here, but it must not be assumed that improvement is easy. There is yet room for considerable study before the final answer is given. The diagrams show that boiler efficiency can be improved by reducing the rate of firing per square foot of grate. If the same power is to be maintained, this means an increase in the size of grate so that the total amount of coal burned may be maintained. Locomotive designs adopted by various railroads in the last few months show development along this line, but as yet no testing-plant analysis of performance is available.

37 In this connection the author calls attention to the fact that in the discussion of boiler efficiency in relation to rate of operation it is usual, as has been done above, to measure the rate of firing in terms of coal per square foot of grate per hour. This assumes that the grate area is the controlling factor in determining the efficiency of combustion. Grate area is an important factor, but not the only one. Firebox volume must also be considered. The test-plant data are not yet sufficiently complete to permit a determination of the relative values of grate area and firebox volume, but one series of tests with high-volatile coal suggests that with this fuel the volume is the more important. The point deserves more study and it may be found that combustion efficiency can be best improved by a slight increase in grate and a considerable increase in firebox volume with appropriate arrangement for giving a long flame-way.

38 The question is extremely complex and can be most adequately answered by further test-plant work. Doubtless this is but one of many problems by the solution of which the locomotive testing plant will advance still further the development of the steam locomotive.

Conclusion

39 For the future we look with confidence for a continued growth in the use of the locomotive testing plant. The great increase in locomotive efficiency which has been seen to have taken place in the last twenty years makes further advance more difficult, and makes exact knowledge a necessary condition for such advance. As it is the function of the locomotive testing plant to provide exact knowledge, it is not surprising that new plants are under

consideration by railroads and by locomotive builders. A very great step forward could be made if the American Railway Association were to construct or to take over a locomotive testing plant to be devoted to the scientific and impartial study of locomotive designs and devices, and to undertake research work concerned with the basic scientific laws governing locomotive operation. Correct locomotive design is based on definite natural laws, knowledge of which can be obtained only by accurate experiment. The great influence which the locomotive testing plant has had on locomotive design is due to the opportunity it has given for thorough and accurate study of the fundamental laws governing locomotive operation. It is evident that the wider and the more accurate the knowledge a designer has regarding the laws governing the operation of his product, the more confidently and the more successfully can he proceed with improvements in design.

APPENDIX NO. 1

40 In the separation of the boiler efficiency into its component factors to produce the diagram in Fig. 3 we have by definition:

Overall boiler efficiency equals heat utilized in evaporation and superheating as percentage of heat in coal fired. Represent this by F

Efficiency of combustion equals heat produced in firebox as percentage of heat in coal fired. Represent this by Fc

Efficiency of heat absorption equals heat taken up by the boiler expressed as percentage of heat produced. The heat taken up by the boiler includes the heat utilized in evaporation and superheating and the heat lost by external radiation, the latter being assumed to be 5 per cent of the former, so that the total heat taken up is 1.05 times the heat utilized. Represent the efficiency of absorption by Fa.

From the foregoing it follows that $F = \frac{Fa}{1.05} \times \frac{Fc}{100}$ or

$$105F = Fa \times Fc \quad \ldots\ldots\ldots\ldots [1]$$

But in the diagrams in Fig. 3 the relation of each of the efficiencies to the rate of firing is represented by a straight line. If G be the rate of firing in pounds of dry coal per square foot of grate per hour, the three efficiencies as plotted have the form —

$$F = m - nG \quad \ldots\ldots\ldots\ldots [2]$$
$$Fa = a - bG \quad \ldots\ldots\ldots\ldots [3]$$
$$Fc = c - dG \quad \ldots\ldots\ldots\ldots [4]$$

41 In the lines plotted the coefficients have the following values:

Locomotive class	m	n	a	b	c	d
K-4s	84.0	0.228	88.0	0.009	100.0	0.263
I-1s	80.0	0.200	85.5	0.015	98.0	0.228

42 A combination of Equations [1], [2], [3], and [4] would give

$$105(m - nG) = (a - bG)(c - dG) = ac - (bc + ad)G + bdG^2 \quad . [5]$$

This cannot be strictly true if a, b, c, d, m and n are constants, but it is approximately true if the term bdG^2 is small compared with the sum of the other terms on the right-hand side of Equation [5]. In

practice this is found to be the case, and the coefficients of the three lines are then connected by the relations

$$105m = ac \quad \ldots\ldots\ldots\ldots [6]$$
$$105n = bc + ad \quad \ldots\ldots\ldots\ldots [7]$$

43 It must be understood that the use of these three straight-line formulas is justified so long as they are understood to be a closely accurate representation of the test results and not as expressing exact basic laws.

APPENDIX NO. 2

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DISCUSSION

A. I. Lipetz.[1] We ought to be very much indebted to the author for his admirable paper which gives, in a very concise form, a complete synopsis of the work done by the American locomotive testing plants. The writer supports his recommendation that the American Railway Locomotive Association build a locomotive testing plant to be devoted to scientific and impartial study of locomotive designs and devices. No doubt such a plant would be of greatest value to the proper development of locomotive designs in this country, and to the matter of the selection of the best devices from the many on the market of railway supplies. It would help to save money wasted on trying various devices on locomotives in service.

However, the writer would like to make several remarks with respect to Mr. Fry's statements, and also take exception to his assertion that the constancy of conditions and accuracy of measurements on a testing plant give the tests on the latter a great advantage over road tests.

To the author's statement that the locomotive testing plant is purely of American origin, the writer would like to add that as far back as 1881 Alexander Borodin, chief engineer of the Russian Southwestern Railway, tested locomotives in stationary condition in order to find out the advantages of compounding and steam jacketing. He blocked the engine in such a way as to raise the main driving wheels, which he used as pulleys. By means of belts the total output of the locomotive was utilized for driving the locomotive machine shops at Kieff. In such a way, he proved by stationary tests that compounding offers a gain of about 15 to 20 per cent in fuel, whereas jacketing is of little value. His assistant,

[1] Consulting Engineer, American Locomotive Company, Schenectady, N. Y. Mem. A.S.M.E.

L. M. Loevy, repeated the tests on road engines, and confirmed Borodin's results. These tests were the beginning of the wide application of compounding to Russian locomotives, which lasted for approximately over 25 years — up to the time when superheating took its place.

While the Borodin tests were not carried out on exactly what could be called a stationary testing plant, the method of testing was the same as that later employed on the American testing plants, with the difference, however, that instead of wasting the work of the locomotive in water brakes, Borodin utilized the performance of locomotives in producing useful work. Therefore, while Mr. Fry's statement may be literally correct, the writer thinks that it requires an amendment in the form of a reference to these first stationary locomotive tests. Mr. Fry is evidently familiar with Borodin's tests, as in Appendix No. 2 to his paper, page 1283, he gives the title of Mr. Borodin's paper published in the Proceedings of the Institution of Mechanical Engineers, London, in 1886.[1]

In regard to Mr. Fry's inability to find records of any real testing plants installed outside of the United States, the writer would like to mention that one testing plant was installed in 1904 in England at the Swindon locomotive shops of the Great Western Railway, which was described by G. J. Churchward in *The Engineer* (London),[2] and two were built in Russia: one at the Putiloff Locomotive Works and another at the Alexandrovsky railway shops, both in Petrograd. The Putiloff testing plant was completed in 1904, and a full description of it was published by its designer, Prof. M. V. Gololoboff, in 1907.[3] The Alexandrovsky testing plant was built only recently. A third Russian locomotive testing plant, which had been built in Esslingen, Germany, was described last year in *The Engineer* (London).[4] The plant was used during 1924 for testing the 2-10-2 Russian 1200-hp. Diesel-electric locomotive in comparison with a 0-10-2 Russian steam locomotive. It has since been dismantled and shipped to Russia for installation as a permanent testing plant. Some particulars concerning the plant are also given by Prof. G. Lomonossoff in his book on the Diesel-electric locomotive.[5]

Mr. Fry states that it was shown at the Purdue plant that the draft-producing action of the exhaust steam was independent of

[1] Even prior to that date, Mr. Borodin published a description of a locomotive testing plant in the *Organ für die Fortschritte des Eisenbahnwesens*, 1881, nos. 4 and 5.

[2] December 22, 1905, p. 621.

[3] Bulletins of the St. Petersburg Institute of Technology (in Russian), vol. XVIII.

[4] November 14, 1924, pp. 553-554.

[5] Die Dieselelektrische Lokomotive, by Prof. G. Lomonossoff, Berlin, 1924, pp. 83-89.

the intermittency of the exhaust, the essential factor in the produced suction being the quantity of exhausted steam. He remarks further that this fact "is often overlooked, particularly when the claim is made that a multi-cylinder locomotive gives a more efficient exhaust action." Dr. Goss' tests at Purdue actually showed that the capacity of the steam jet through the exhaust nozzle as a means of producing draft is nearly proportional to the weight of steam discharged per unit of time,[1] and that with a given weight of steam discharged, whether in the heavy exhausts incident to slow speed, or in the more rapid impulses which are sent forth at higher speeds, the draft is practically the same.[2] This would tend to indicate that the *average*[3] vacuum is not affected by the number of impulses, but this is not the claim which is usually made regarding multi-cylinder locomotives and, to be more specific, the three-cylinder locomotives. The claim is that the more frequent and consequently lighter exhausts give, at the same average vacuum, a more uniform draft, resulting in a better efficiency of the boiler as indicated by the greater evaporation per pound of fuel. The reason is obvious and lies in the fact that the heavy individual impulses cause incomplete combustion and entail great cinder and spark losses, in accordance with other tests made by Dr. Goss, whereas the more numerous and lighter impulses, giving the same average draft action, do not cause any such losses. This fact has been known since the first two-cylinder compound locomotives were built in the early eighties. These locomotives — with only two exhausts per revolution — showed a lower boiler efficiency than the two-cylinder simple-expansion locomotives with four exhausts. Von Borries, the well-known authority on compound locomotives and their advocate, considered this fact as the most important disadvantage of the two-cylinder compound locomotives.[4] Tests made later in Russia fully confirmed the fact that two exhausts give a lower boiler efficiency than four.[5]

It is true that Dr. Goss also said that "variations in the character of the exhaust jet — such as result from changes in speed or cut-off — do not in themselves affect the efficiency of the boiler,"[6] but out of the 35 tests on which this statement was based, only three were made at a rate of combustion of over 150 lb. per sq. ft. of grate area per hour, and the coal fired during the tests was of

[1] Locomotive Performance, by Dr. W. F. M. Goss, New York, 1907, p. 138.

[2] *Ibid.*, p. 238.

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[4] Von Borries. Compound Lokomotiven. *Organ für die Fortschritte des Eisenbahnwesens*, 1883, pp. 190-192.

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[6] Locomotive Performance, by Dr. W. F. M. Goss, New York, 1907, p. 146.

such quality that even at a very high rate of combustion, namely, 241 lb. per sq. ft. per hr., the spark losses amounted to only 15.5 per cent.[1] The results would have been different if lighter coal had been used.

The fact that four exhausts give a lower boiler efficiency than six has been proved in this country and in England with the three-cylinder locomotives, as can be seen from the official test figures of the 4-8-2 three-cylinder locomotive No. 5000 on the Lehigh Valley Railroad. The overall boiler efficiency was 77.58 per cent at an hourly dry-coal rate of firing of 69.3 lb. per sq. ft. of grate area, whereas at the same rate of firing the efficiencies obtained from tests with Pennsylvania Railroad locomotives at the Altoona testing plant never exceeded 72 per cent.

Likewise, tests with the 2-8-2 Missouri Pacific three-cylinder locomotive made at the Altoona testing plant give a boiler efficiency curve which is higher than those of all the Pennsylvania locomotives tested at Altoona, except one class — the K4s (4-6-2 two-cylinder passenger) locomotive, which shows an exceptionally good boiler performance, probably due to the use of the kind of fuel best suited to the actual boiler proportions.

The better efficiency of the boiler on three-cylinder locomotives is also illustrated by the fact that the economy in fuel on three-cylinder locomotives is very often higher than that in water; this can be explained by the more complete combustion and smaller cinder losses. Tests made on the London & North-Eastern Railway of England showed that on three-cylinder, Atlantic-type locomotives the economy in fuel was from 19 to 25 per cent, at the time when economy in water was only from 10 to 18 per cent.[2]

As regards four-cylinder locomotives, there is no difference between them and two-cylinder locomotives, because with cranks at 90 and 180 deg. two exhausts always coincide in time, and the number of exhausts on two-cylinder and four-cylinder locomotives is the same.

The exception which the writer takes to Mr. Fry's statement is the preference which he gives to stationary tests as compared with road tests. The essential part of the stationary test is the constancy of conditions, such as speed, cut-off, etc.; therefore, if we could test locomotives on the roads under constant conditions, there would be very little difference between such tests and those on the testing plant. Moreover an advantage would be gained because the locomotive would be tested with its tender, which usually is

[1] Locomotive Performance, by Dr. W. F. M. Goss, New York, 1907, p. 162.

[2] The Three-Cylinder, High-Pressure Locomotive, by H. N. Gresley; advance copy of a paper read before the Institution of Mechanical Engineers, London, on July 7, 1925, p. 8. See also pp. 18 and 19 regarding the uniformity of draft as measured by a draft recorder of the Cambridge Instrument Co.

taken off for laboratory tests, and under actual conditions of running on a track where the vibrations of movements affect the efficiency of evaporation. Some roads have sufficiently long grades, or level tracks, which would offer the possibility of testing locomotives under constant conditions. For instance, there is on one road in this country a division of about 38 miles in length, consisting practically of a uniform grade of 1.8 per cent, and an adjacent division of a uniform grade of 2.2 per cent of almost the same length. A freight locomotive with a train could be tested at a uniform speed with constant cut-off and throttle opening, temperature, boiler pressure, etc., during a period of two to three hours on each division, and a passenger locomotive during the same period of time on both divisions together. This would be just as convenient as testing a locomotive at a stationary plant. If a dynamometer car were attached to the tender of the locomotive, the total output of the locomotive could be registered just as accurately as on a testing plant, and weather conditions would have no influence, as the resistance due to temperatures and wind would be included in the readings of the dynamometer. As regards accuracy, there would be no difference between the readings in a dynamometer car and those on a stationary plant. Even gas analyses could be made, and temperatures in the firebox and other parts of the boiler could be measured if the necessary apparatus and galvanometers with compensation devices were installed in the car. Tests of this kind are being performed in Europe as a matter of routine. In two countries, Russia and Germany, special test departments attached to the Central Railways Administration have been in existence for years. Practically every year since 1898 such tests have been going on in Russia. It is interesting to note that many of the results which were obtained on the Altoona testing plant have been either confirmed, or even preceded by the Russian tests. So, for instance, the length of tubes in freight and passenger locomotives built in Russia since 1901 was made 4660 mm. at 46 mm. inside diameter (2-in. tubes); this would give a ratio of 101 — in striking conformity with Bulletin No. 21 of the Pennsylvania Testing Plant published in 1913, which recommends a ratio of 102. It has been customary in western Europe and in Russia to design superheater locomotives with cut-offs of less than 50 per cent, this on account of results of road tests made in the first decade of the present century. The difference between this practice and that of the Pennsylvania Railroad is that in the European locomotives the 85 per cent cut-off for starting is being obtained by the ordinary Walschaerts gear, whereas in the Pennsylvania 50 per cent cut-off Decapods special provisions are made in the valves for starting. Any attempt on the part of the engineer on a Russian superheated locomotive to use cut-offs running over 50 per cent would result in slipping of the driving wheels. The relation between water consumption per horsepower, cut-offs,

and speeds was fully established for all Russian compound and simple-expansion locomotives, and the most economical combinations of the throttle opening and cut-offs were studied, especially for compound locomotives, where they are more important than for locomotives with simple expansion.

In order to check up whether the road-test figures are of real practical use, the following method has been employed in Russia. From the tractive-effort curves a very exact graphical speed-distance chart is drawn for a train of a certain tonnage. The most economical combinations of throttle opening and cut-off are chosen for each portion of track in accordance with speed. Further, the elements of work due to resistance and acceleration are calculated for the same portions of track (also graphically) and the consumption of fuel and water is thus ascertained. Then a test is made for which the driver is instructed to employ certain combinations, and a verification of speed, fuel, and water consumption is made. The actual figures do not differ usually more than by from 3 to 5 per cent.

Even the fact that the firebox volume must be considered in addition to the firebox heating surface and the grate area, is known to European locomotive designers. The writer would mention here that the reason for the high location of boilers on some European locomotives, where the diameter of the boiler barrel does not require such high location, is due mostly to the desire to give the firebox the proper depth and volume.

The writer has no intention of criticizing Mr. Fry's paper, or contesting his conclusions in so far as they refer to the testing plants, but he wishes to correct the impression which one might obtain from the third paragraph of the paper, namely, that laboratory tests are the only means of studying locomotive performance. Many roads that are unfortunate in having long grades, or fortunate in having long level stretches, can obtain as reliable data from road tests as one can get on a locomotive testing plant. Of course, it must be remembered that road tests made on a division with variable profile are very misleading. Average figures can have only comparative value with locomotives tested on the same division. If the relations between variables of locomotive performance could be represented by straight lines, this method might have been correct, but owing to the fact that curves of fuel and steam consumption per unit of work, of draft in the smokebox, etc., have a maximum (or minimum) about the average abscissas, the average figure from a test made on a variable profile has no meaning.

In conclusion, Mr. Fry's statement that the American testing plants materially influence locomotive designs abroad is especially correct with respect to the Purdue plant, which exerted a tremendous influence on European locomotive design as reflected in

European locomotive literature, where references to Dr. Goss' tests abound.

GEORG LOMONOSSOFF.[1] The author indicates that locomotive testing plants are of purely American origin and that he has been unable to find a record of any real testing plant installed outside of the United States.

The writer is very far from denying the significance of American achievements in the field of locomotive laboratory tests and espe-

FIG. 4 LOCOMOTIVE TESTING PLANT IN DÜSSELDORF, GERMANY, INSTALLED FOR TESTING LOMONOSSOFF DIESEL-ELECTRIC LOCOMOTIVES

cially the remarkable work of his friend Dr. W. F. M. Goss, but for the sake of truth he permits himself to call attention to the fact that the first locomotive laboratory was installed in Kieff in 1882 by Alexander Borodin. This laboratory and the classical tests carried out there by Mr. Borodin were described in the Proceedings of the Institution of Mechanical Engineers, London, 1886, p. 317.

The laboratory in Kieff was of a temporary character and was abandoned in 1886. But in 1906 another was erected in Leningrad

[1] Berlin, Germany.

(then St. Petersburg) at the Putiloff Works as a permanent plant on six coupled wheels. Russian railway science is indebted for this to the engineers Smirnoff and Gololoboff. The latter carried out on this plant during the years 1906-1914 numerous tests, some of which have been published in Russian.

In 1912 it became evident that the brakes of the Putiloff plant were not capable of handling modern locomotives, and the following year the Russian Government instituted a new plant at the Alexander Works (now the Proletarian Works) near Leningrad. This plant was intended to serve not only the needs of the Russian railroads but also the pedagogical purposes of the Institute of Ways of Communication. War and revolution, however, delayed its building, and it was not until 1924 that the testing plant began its activities. At the present time tests are being carried out there by Professor Grinenko.

An order from Germany for two Diesel locomotives compelled the writer to erect a temporary Russian testing plant in Esslingen, which was later removed to Düsseldorf. This plant is described in German in his book Die Dieselelektrische Lokomotive (1924), pp. 83-87, in his article in the *Organ für die Fortschritte des Eisenbahnwesens*, 1924, pp. 166-170 and in English in an article in *The Engineer* of November 14, 1924, p. 553. The accompanying photograph (Fig. 4) shows the present state of this testing plant, as well as the writer's second Diesel locomotive.

In conclusion the writer is compelled to state that with all the conveniences which locomotive testing plants offer they are not able to replace road tests. This does not refer to trial runs of the usual type, but to scientific tests on sections of a constant profile. This matter has been discussed in the above-mentioned article in the *Organ für die Fortschritte des Eisenbahnwesens*, and in the writer's book Lokomotivversuche in Russland, published in 1926 by the Verein deutscher Ingenieure, as well as in numerous articles contributed by him to Russian periodicals.

THE AUTHOR. Mr. Lipetz questions four statements made in the paper:

(*a*) That the locomotive testing plant is of purely American origin. This statement was made with full knowledge that Borodin in 1881 had jacked up locomotives, fitted belts to the driving wheels, and made tests under these conditions. This ingenious but crude method enabled certain tests to be made, but the arrangement could hardly be called a locomotive testing plant. The plant at Purdue was the first to embody the principles which have been so fruitful in the production of knowledge, and we have Dr. Goss's statement that the Purdue plant was installed before he had any knowledge of Borodin's work.

(*b*) That no real testing plants have been installed outside of the United States. It is true that a set of testing rolls was installed

at Swindon. The record of work done on these rolls, however, is not sufficient to justify their being described as a real locomotive testing plant. The three Russian plants were overlooked. Thanks are due to Mr. Lipetz for completing the record.

(*c*) That the efficiency of the draft-producing action of the exhaust is independent of its intermittency. This question can be settled more easily and much more definitely than by examining the various comparative tests quoted by Mr. Lipetz. In any series of tests made with a given boiler it will be found that for a given quantity of steam produced and exhausted the boiler efficiency will be the same whether the steam be exhausted at 40 r.p.m. or at 140 r.p.m. That is to say, the boiler efficiency is determined by the rate at which steam is produced and exhausted, and is not affected by the intermittency of the exhaust.

(*d*) That tests can be made on a locomotive testing plant better than on the road. To any one at all familiar with the details of locomotive testing Mr. Lipetz' objection to this statement will be surprising, and a refutation will seem unnecessary. Very few railroads have a conveniently situated track on which it would be possible to run an hour's test at 60 miles per hour at absolutely uniform speed and cut-off. In a road test it is practically impossible to arrange in advance the exact speed and cut-off at which a test shall be run. With the cut-off determined, the speed will be determined by the equipment available and the atmospheric conditions. This makes it difficult to duplicate exactly any given road test. When in addition the convenience of working with instruments stationary and in close proximity to laboratory and computing room is considered, it seems to the author that for accuracy and convenience, tests on a locomotive testing plant are far ahead of road tests of locomotives.

In reply to Professor Lomonossoff, the author desires to express appreciation for the information given regarding the Russian locomotive testing plants. The author agrees with Professor Lomonossoff that laboratory tests will never replace road tests of locomotives. There is a certain amount of working involving drawbar pull and train resistance which is necessarily carried out on the road. He is still of the opinion, however, and he has much well-informed authority with him on this, that for accurate and convenient measurements of boiler conditions and of steam consumption in relation to cut-off and speed, the testing plant is very much to be preferred.

No. 1996

DUTY TESTS OF VERTICAL TRIPLE-EXPANSION PUMPING ENGINES, MILWAUKEE, WIS.

RESULTS OBTAINED AT RIVERSIDE PUMPING STATION

By Charles A. Cahill,[1] Milwaukee, Wis.

Member of the Society

Milwaukee, Wisconsin, has equipped its new Riverside Pumping Station with three vertical triple-expansion pumping engines. The reason for the choice of engines of this type was their great reliability, high economy, and low cost of maintenance. The author believes that the units A and B recently tested have shown a duty record that has not been equaled to date by any using steam as a working medium for pumping water.

THE New Riverside Pumping Station was placed in operation with one unit on July 1, 1924, the second unit going into service in October. The two units in operation are known as "high-service" engines and operate against a head of 280 ft., while the third unit is for a head of 170 ft. and will pump into the Kilbourn Park Reservoir as soon as it is connected to the discharge line. This station was built to place six engines in service, three of which are now installed.

2 The city's growth is toward the high-service district and the low-service unit can be adapted for permanent operation on the high-service distribution by simply changing the pump plungers, neck rings, and stuffing boxes, the pump ends being constructed of ample strength for this service.

3 Milwaukee is the birthplace of the vertical triple-expansion pumping engine. The first engine was placed in service in the Tenth Street High-Service Station in 1886 and is ready to take its share of the load today, if required.

4 Milwaukee's first pumping engines in a municipally operated water works were placed in operation in 1874 and consisted of two 8,000,000-gal. engines of the Woolf compound duplex beam

[1] Cons. Engr., Cahill & Douglas.

Contributed by the Milwaukee Spring Meeting Committee and presented at the Spring Meeting, Milwaukee, Wis., May 18 to 21, 1925, of The American Society of Mechanical Engineers.

type, with bucket and plunger pumps. Steam was supplied at a working pressure of 50 lb. per sq. in. from Cornish boilers. A duty of 76,955,000 ft-lb. was obtained per 100 lb. of anthracite

FIG. 1 RIVERSIDE PUMPING STATION, MILWAUKEE, WIS.

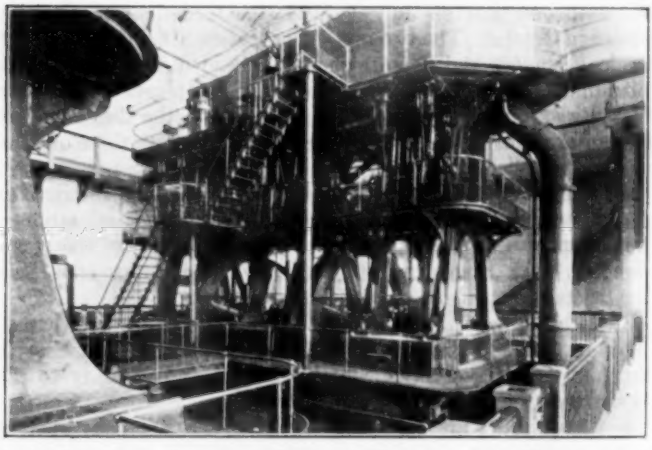

FIG. 2 RIVERSIDE PUMPING STATION, ENGINE A AS VIEWED FROM MAIN GALLERY

costing $7.60 per ton of 2000 lb. These engines were dismantled in 1913. They were built by the E. P. Allis Company, of Milwaukee. The late Thomas McMillen erected and started these engines, accepted the position of chief engineer of the Milwaukee Water Works on October 1, 1873, and held this position con-

tinuously until his death on May 1, 1920, a record almost without a parallel in municipal service.

5 In 1882 a 12,000,000-gal. pumping engine was furnished the city by the E. P. Allis Company. This engine was a vertical tandem compound beam type with bucket and plunger pump, the two steam cylinders and pump being vertically in line with the low-pressure cylinder above the high-pressure. Motion was transmitted to the flywheel through a beam. It is believed that this design was never repeated. It showed a duty of 104,802,431 ft-lb. per 100 lb. of coal.

6 In 1891 the city purchased from the E. P. Allis Company for the North Point Pumping Station an 18,000,000-gal. vertical triple-

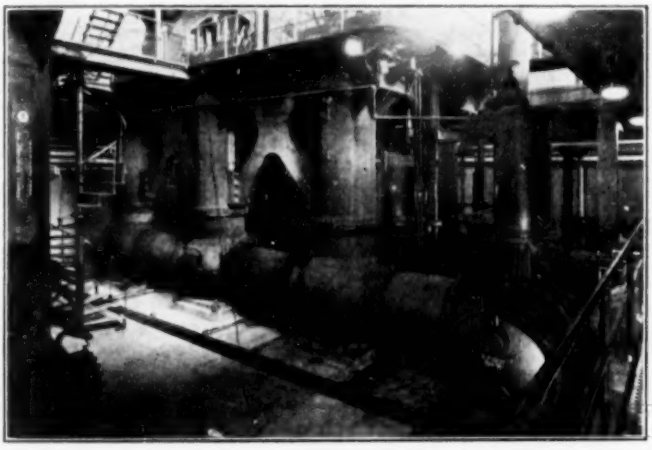

FIG. 3 RIVERSIDE PUMPING STATION, PUMPS OF ENGINE A

expansion pumping engine. This unit held the world's record for economy for some time. After 30 years of continuous service it has been thoroughly overhauled at a very nominal cost and will be placed in service for another 30 years' run, or until such advances are made in economical pumping machinery that it will not be profitable to operate it.

7 It is thus seen that since 1886 all pumping engines for waterworks service in Milwaukee have been of the vertical triple-expansion type, and all except two have been built by the E. P. Allis Company or its successor, the Allis-Chalmers Manufacturing Company. They are a home product.

8 Steam pressures in Milwaukee pumping stations have gone from saturated steam at 50 lb. per sq. in. in 1874 to 210 lb. and 125 deg. superheat at the Riverside Station. In this respect our pumping stations have followed the universal rise in steam pres-

sures and temperatures during the recent years, but have not as yet reached the so-called "super-pressures."

9 Milwaukee has increased its pumping capacity from time to time with this type of pumping engine for the following simple reasons:

(*a*) *Extreme Reliability.* In the 39 years' operation of this type of pumping engine there has never been any anxiety on the part

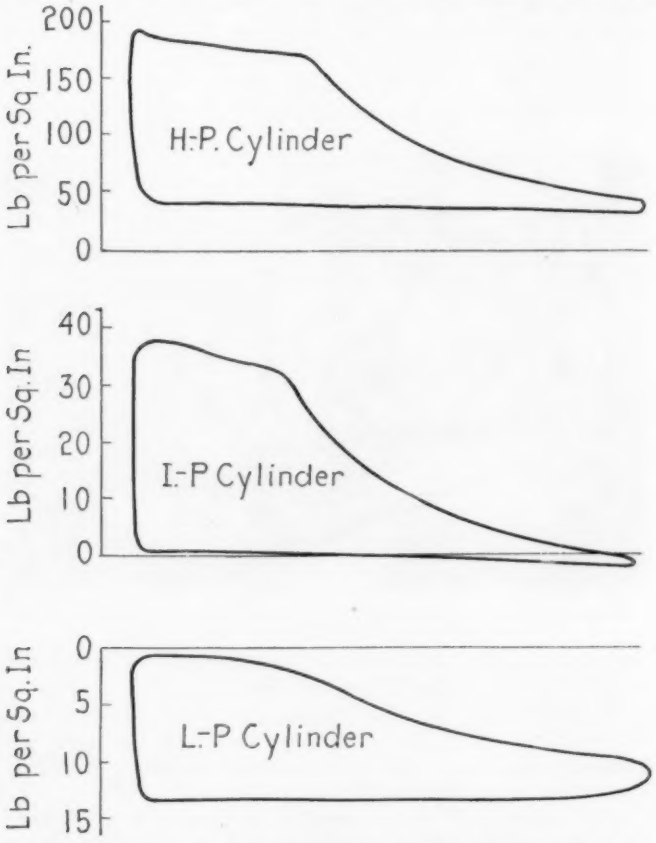

FIG. 4 TYPICAL HEAD-END INDICATOR CARDS, OFFICIAL TEST OF ENGINE A, RIVERSIDE PUMPING STATION

of the water-works' officials or their staff as to its ability to do all it was guaranteed to do and much more. There has never been a breakdown.

(*b*) *High Economy.* The triple-expansion pumping engines at North Point have shown over a long period of years a remarkable

high station duty over a wide range of capacity. They have also maintained this high duty over a long period of years.

(*c*) *Low Maintenance Cost.* The maintenance of this type of pumping equipment has been extraordinarily low in cost. Note the general repairs on the original 18,000,000-gal. unit at the North Point after a continuous run of 32 years.

10 The pumping engines at the Riverside Station are equipped with bleeder feedwater heaters taking steam from the low-pressure receiver. These heaters were not in use during the acceptance tests, the guarantees being made with exhaust heaters in service only. The bleeder feedwater heaters will increase the thermal efficiency of the units from 6.5 to 7 per cent. The results of the

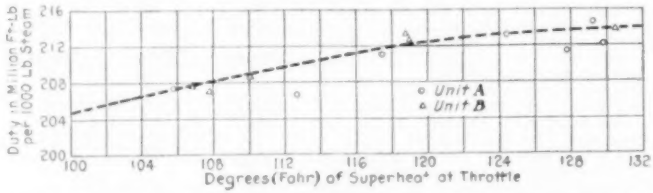

FIG. 5 EFFECT OF SUPERHEAT ON ECONOMY OF VERTICAL TRIPLE-EXPANSION ENGINES, UNITS A AND B, RIVERSIDE PUMPING STATION

acceptance tests on the Riverside units A and B are given in detail in Table 2, and the principal dimensions of these units in Table 1. The author believes from the acceptance tests on engines A and B at Riverside that this duty record has never been equaled by any steam-driven pumping engine to date.

11 The author believes that the continual increase in the thermal efficiency of this type of pumping engine is due to the very careful refinement of details and, of course, to the use of higher pressures and temperatures, and also to the fact that all engines of this type are invariably given a careful duty test.

12 It is interesting to note that the 18,000,000-gal. unit No. 4 at North Point Station set a new record in economy in 1892, and the results of recent tests at Riverside lead to the belief that after an interval of 33 years another record has been established with the same type of engine built by practically the same company.

TABLE 1 PRINCIPAL DIMENSIONS OF THE PUMPING ENGINES INSTALLED IN 1924 AT THE RIVERSIDE PUMPING STATION, MILWAUKEE, WIS.
(Builder, Allis-Chalmers Mfg. Co., West Allis, Wis.)

		Engine A	Engine B	Engine C
1	Number of steam cylinders	3	3	3
2	Diameter of steam cylinders, in.	32-58-96	32-58-96	26½-48-80
3	Diameter of piston rods, in.	6½-6½-8	6½-6½-8	6½-6½-8
4	Stroke of steam piston and water plungers, in.	60	60	60
5	Area of h. p. piston, head end, sq. in.	804.25	804.25	551.55
6	Area of i. p. piston, head end, sq. in.	2642.09	2642.09	1809.56
7	Area of l. p. piston, head end, sq. in.	7238.25	7238.25	5026.56
8	Net area h. p. piston, crank end, sq. in.	768.47	768.47	515.77
9	Net area i. p. piston, crank end, sq. in.	2606.31	2606.31	1773.78
10	Net area l. p. piston, crank end, sq. in.	7187.98	7187.98	4976.29
11	Proportion of gross area of pistons, head end	1:3.19:9	1:3.19:9	1:3.28:9.1
12	Proportion of net area of pistons, crank end	1:3.39:9.35	1:3.39:9.35	1:3.44:9.65
13	Number of water plungers (single-acting).	3	3	3
14	Diameter of each water plunger, in.	32¼	32¼	35
15	Area of each plunger, sq. in.	848.83	848.83	962.12
16	Displacement of 3 plungers per revolution, gal.	661.43	661.43	749.70
17	Number of valve chambers	6	6	6
18	Number of valves in each cage	35	35	35
19	Number of valve cages in each deck	7	7	7
20	Diameter of valve openings, in.	3⅜	3⅜	3⅜
21	Free valve area per valve, sq. in.	5.5	5.5	5.5
22	Net area of waterway through deck, sq. in.	1348	1348	1348
23	Ratio of net valve area to plunger area	1.59	1.59	1.40
24	Diameter of connected-air-pump bucket, in.	30	30	30
25	Stroke of connected air pump, in.	60	60	60
26	Type of condensers (surface)	Water-works	Water-works	Water-works
27	Number of condensers	1	1	1
28	Diameter of condenser tubes, in.	1¼	1¼	1¼
29	Inside diameter of condenser shell, in.	63	63	63
30	Cooling surface in condenser, sq. ft.	1500	1500	1200
31	Diameter of attached feed pump, in.	2½	2½	2½
32	Diameter of attached air compressor, in.	2½	2½	2½
33	Size of primary closed heater, sq. ft.	250	250	250
34	Shipping weight of unit complete, lb.	1,630,000	1,630,000	1,560,000
35	Weight of single heaviest piece, lb.	45,000	45,000	45,000
36	Revolutions per minute at rated capacity.	23.5	23.5	23.5
37	Rated capacity, million gal. per 24 hr.	22	22	25
38	Total head for which unit is designed, ft.	280	280	170
39	Plunger speed at rated capacity, ft. per min.	235	235	235
40	Diameter of main journals, in.	20	20	20
41	Length of main journals, in.	30	30	30
42	Diameter of h. p. and l. p. crankpins, in.	12	12	12
43	Length of h. p. and l. p. crankpins, in.	10	10	10
44	Diameter of i. p. crankpins, in.	18	18	18
45	Length of i. p. crankpins, in.	10	10	10
46	Diameter of crosshead pins, in.	11	11	11
47	Length of crosshead pins, in.	12	12	12
48	Diameter of plunger rods, in.	5⅛	5⅛	5⅛
49	Diameter of flywheel, ft.	18	18	18
50	Weight of each flywheel, lb.	65,000	65,000	55,000
51	Inside diameter of plunger barrel, in.	62	62	62
52	Thickness of metal in plunger barrels, in.	2½	2½	2½
53	Thickness of metal of discharge chambers, in.	2½	2½	2½
54	Inside diameter of suction chambers, in.	71½	71½	71½
55	Thickness of metal in suction chambers, in.	2½	2½	2½
56	Inside diameter of discharge chambers, in.	71½	71½	71½
57	Width of valve decks, in.	15	15	15
58	Diameter of suction nozzle, in.	42	42	42
59	Diameter of discharge nozzle, in.	42	42	42
60	Velocity through suction nozzle at rated capacity, ft. per sec.	3.53	3.53	4.02
61	Velocity through discharge nozzle at rated capacity, ft. per sec.	3.53	3.53	4.02

TABLE 2 SUMMARY OF OFFICIAL TESTS OF PUMPS A AND B, RIVERSIDE PUMPING STATION, MILWAUKEE WATERWORKS [1]

(Object of Test: To determine fulfillment of duty guarantees.)

	Contract Conditions	Pump A	Pump B
1	Capacity, million gal. per 24 hr.	22	22
2	Total head in feet	280	280
3	Steam pressure at throttle, lb. per sq. in.	200	200
4	Speed in r.p.m.	23.5	23.5
5	Duty in million ft-lb. per 1000 lb. steam	200	200
6	Date of test (1925)	Jan. 13	Jan. 27
7	Duration in hours	10	10
	Average Steam Pressures		
8	Barometer in inches of mercury	29.61	30.07
9	Barometer in lb. per sq. in.	14.51	14.73
10	Steam pressure at throttle, lb. per sq. in. (gage)	199.02	200.50
11	Steam pressure at throttle, lb. per sq. in. (absolute)	213.53	215.23
12	Steam pressure in 1st receiver, lb. per sq. in. (gage)	35.70	35.93
13	Steam pressure in 2d receiver, lb. per sq. in. (gage)	—1.12	—1.06
14	Steam pressure in 2d receiver, lb. per sq. in. (absolute)	13.39	13.67
15	Vacuum in condenser in inches of mercury	28.58	28.93
16	Vacuum in condenser referred to 29.92-in. barometer	28.89	28.78
17	Absolute pressure in condenser, lb. per sq. in.	0.505	0.559
18	Absolute pressure corresponding to exhaust temperature, lb. per sq. in.	0.533	0.534
	Average Temperatures in Deg. Fahr.		
19	Steam at throttle	516.6	506.9
20	Saturated steam at test pressure	387.4	388.1
21	Superheat at throttle	129.2	118.8
22	First receiver	283.0	285.76
23	Condensate from air pump	38.2	38.02
24	Condensate after passing exhaust heater	76.0	77.63
25	Lake Michigan water at suction nozzle	36.2	36.02
26	Jacket and drain water (after entering tank)	189.7	189.0
27	Engine room, main floor	72.0	72.0
28	Exhaust steam (exhaust-pipe, top condenser)	81.7	81.73
	Water Pressures and Heads		
29	Average discharge pressure, lb. per sq. in.	117.13	115.73
30	Average suction lift, ft.	0.94	0.89
31	Differences in elevation, center to center of gages, ft.	3.61	3.66
32	Total head pumped against, ft.	121.68	120.19
	Total Quantities during Test		
33	Total condensate from air pump, lb.	90949	89759
34	Total jacket and receiver drains, lb.	10061	11381
35	Total steam used during test, lb.	101610	101140
36	Per cent of total steam used by jacket and drain	10.5	11.26
37	Total revolutions of engine	14071	14095
38	Water pumped per revolution, gal.	661.43	661.43
39	Total water pumped, million gal.	9.307	9.321
	Average Quantities during Test (Capacity)		
40	Average r.p.m.	23.45	23.49
41	Average piston speed, ft. per min.	234.5	234.9
42	Average rate of pumpage, million gal. per 24 hr.	22.335	22.365
43	Average rate of pumpage in per cent of normal	101.5	101.6
	Power		
44	Total work done during test, million ft-lb.	21,890	21,570
45	Average water horsepower from test data	1101.01	1089.38
46	Average indicated horsepower from cards	1121.88	1128.89
47	Average combined mechanical efficiency, per cent	98.14	96.5
48	Per cent of total i.hp. developed in h. p. cylinder	39.2	40.7
49	Per cent of total i.hp. developed in l. p. cylinder	28.7	28.0
50	Per cent of total i.hp. developed in l. p. cylinder	32.1	31.3

[1] All readings of gages, thermometers, etc., were taken at fifteen-minute intervals, during the tests, with great accuracy. All gages were calibrated before and after tests by means of dead-weight testers. Measurements of condensate and jacket drains was recorded on tested platform scales and carefully checked from hour to hour. Office calculations were carefully made and checked by the Allis-Chalmers Mfg. Co. engineers, working in conjunction with the city's engineers.

(*Continued on page 1302*)

TABLE 2—CONTINUED

		Pump A	Pump B
Hourly Quantities (Average)			
51	Revolutions per hour	1407.1	1409.5
52	Total steam per hour, lb	10161	10114
Heat Data			
53	Total heat per lb. at throttle, B.t.u.	1273.8	1268.7
54	Gross heat supplied in test above 32 deg. fahr., million B.t.u.	129.431	128.316
55	Heat returned per lb. of jacket and drain water above 32 deg. fahr., B.t.u.	157.7	157.0
56	Total heat returned in jacket and drain water above 32 deg. fahr., million B.t.u.	1.681	1.787
57	Heat returned per lb. of condensate leaving exhaust heater, B.t.u.	44	45.6
58	Total heat returned in condensate, million B.t.u.	4.002	4.093
59	Total heat returned in jacket and drain water and condensate, million B.t.u.	5.683	5.880
60	Net total heat supplied during test, million B.t.u.	123.748	122.436
61	Net total heat supplied per lb. steam in test, B.t.u.	1217.9	1210.6
62	Heat available on Rankine cycle based on exhaust temperature, B.t.u.	398.3	396.3
63	Heat available on Rankine cycle under contract conditions based on exhaust temperature during test, B.t.u.	391.9	391.9
Economy As Measured			
64	Duty per 1000 lb. of steam during test in million ft-lb.	214.55	213.27
65	Duty per million B.t.u. furnished, including heat returned in feedwater, million ft-lb.	176.16	176.17
66	Thermal efficiency of entire unit, per cent	22.65	22.65
67	Engine efficiency based on delivered work, per cent	69.1	69.2
68	Steam supplied per delivered hp-hr., lb	9.23	9.29
69	Steam supplied per i.hp-hr., lb	9.05	8.97
70	Net heat supplied per delivered hp-hr., B.t.u.	11,240	11,239
71	Net heat supplied per i.hp-hr., B.t.u.	11,030	10,846
	CONTRACT CONDITIONS		
Economy Based on Correction Curve			
72	Duty per 1000 lb. of steam with steam at 200 lb. gage and 100 deg. fahr. superheat, million ft-lb.	204.6	204.6
73	Guaranteed duty per 1000 lb. of steam as per contract, million ft-lb.	200.0	200.0
Economy Corrected as per Heat Available on Rankine Cycle			
74	Duty per 1000 lb. of steam corrected for heat available on Rankine cycle, million ft.lb.	211.0	210.5

DISCUSSION

GEO. H. BARRUS.[1] The writer wishes to concur with the statement that the superior economy realized was due "to the use of higher pressures and temperatures." Indeed, when compared with engines built by the same shops more than 30 years ago when saturated steam at 125 lb. pressure was used, it appears to be due altogether to superheating and increased pressure, and not at all to "the very careful refinement of details" mentioned, or to any special merit of the engine itself.

From the chart of Fig. 4, the actual effect of superheating in this case can be computed; and the writer finds that the 129 degrees of superheat in engine A added 51.6 million ft-lb. to the duty per 1000 lb. of steam. With saturated steam, therefore, the duty of unit A would have been $214.5 - 51.6 = 162.9$ million ft-lb. per

[1] Expert and Consulting Steam Engineer, Boston, Mass. Mem. A.S.M.E.

1000 lb. of steam, or, proportionately, 133.6 million ft-lb. per million B.t.u.

From the indicator cards given, the effect of the increase of pressure on the duty above what it would have been at 125 lb. can be readily computed. The writer estimates the increase in power due to the increased pressure to be not less than 10 per cent. In other words, at 125 lb. pressure the duty would have been reduced one-eleventh, bringing it down to $133.6 - 12.1 = 121.5$ ft-lb. per million B.t.u.

In 1895 the writer conducted continuous duty trials on an engine of identically the same mechanical type and make as the one under consideration, having a capacity of 24 million gallons in 24 hours, which used saturated steam at 125 lb. pressure. The average duty for the whole run was 129.7 million ft-lb. per million B.t.u. It appears from this comparison that so far as the merits of the engine itself are concerned, its thermal efficiency has decreased instead of increased as averred by the author.

The writer refers also to superior duties attained on other engines using saturated steam, so long ago as 1900. Two Boston engines each gave 156 million ft-lb. per million B.t.u., and one St. Louis engine reached 158 million ft-lb.

ROBERT W. ANGUS.[1] The engines described in this paper are so nearly the same size as the one installed in Toronto in 1906 that the writer compared the results which he obtained in a test made March 26, 1906, with those given in the present paper.

The Toronto engine has steam cylinders 32, 60, and 90 in. diameter by 60 in. stroke, with single-acting water plungers each $32\frac{1}{4}$ in. diameter, while the Milwaukee engines have corresponding dimensions of 32, 58, and 96 in. by 60 in. with $32\frac{7}{8}$ in. plungers.

The test speed of the Toronto engine was 19.83 r.p.m.; that of the Milwaukee engine A being 23.45; and the net water pressure in the Toronto engine was 103.6 lb. per sq. in. as compared with 121.68 lb. per sq. in. for engine A. The Toronto machine was supplied with steam at 148.3 lb. with 1.22 per cent moisture. While the duty obtained in the Toronto engine was 172.3 millions of foot-pounds per thousand pounds of steam used, the efficiency of the engine referred to the Rankine cycle was 69.2 per cent as compared with 69.1 per cent in engine A as given in item 67. The two machines have therefore given the same efficiency when compared on the same basis, and it would look as if this type of machine had reached its ultimate efficiency some years ago.

The writer calls attention to the very high mechanical efficiency of 98.14 per cent on engine A and asks whether the indicator

[1] Professor of Mechanical Engineering, University of Toronto, Toronto, Can. Mem. A.S.M.E.

springs were carefully calibrated and what effort was made to see that this figure represented correctly the true result.

While other water-works plants are largely adopting the centrifugal pump, Milwaukee still adheres to the older type, and it is not clear why this has been done. It is true that higher duties have been obtained with vertical machines than with the centrifugal pumps, but the latter are being improved continually, and in a recent purchase by Toronto a duty of 157.5 million ft-lb. per thousand pounds of steam was guaranteed with saturated steam at 150 lb. per sq. in., and the mechanical efficiency of centrifugal pumps of the capacity of the one described in this paper has reached 87 per cent. The steam-turbine-driven unit is better suited to the use of superheated steam than the vertical engine.

Against the lower duty must be put the lower first cost of the centrifugal unit, and the writer also believes that the maintenance cost of the latter unit is lower than that of the vertical machine. The pumps described have nearly 1500 valves, and the renewal of these is very expensive; and the other parts of the large machines require them to be kept in good condition to maintain their efficiency. In the centrifugal pump the only parts that ordinarily have to be renewed are the impeller and rings, and in over ten years' service these have shown no depreciation in Toronto. The cost of one of these impellers is approximately $1200 (not including customs duty), which would not go very far in the way of renewing the valves, etc., which would be necessary in the maintenance of the larger machine.

LORAN D. GAYTON.[1] It is safe to predict that this paper is a description of what will probably be the last large triple-expansion pumping-engine station to be erected in this country. In Chicago we have many engines of this type. Two years ago, in designing a new plant, an exhaustive comparison was made of all types of pumping equipment, resulting in the choice of turbine-driven centrifugal pumps.

The writer is informed that the Milwaukee units cost $250,000 each. In Chicago an 80 m.g.d. turbine-driven centrifugal pumping unit for a total head of 150 ft. with a guaranteed duty of 200 million foot-pounds of work per thousand pounds of steam (300 lb. gage pressure and 200 deg. fahr. superheat) was purchased for about $86,000.

In our particular case the buildings for the triple-expansion engines would cost $294,000 more than for turbine-driven centrifugals. There is no doubt about the efficiency of the triple-expansion pumping engine, but it is not enough greater than that of the turbine-driven centrifugal to offset the greater cost of equipment and buildings.

[1] Engineer of Water Works Design, Chicago, Ill. Mem. A.S.M.E.

THE AUTHOR. In answer to a question[1] the author would say that the steam-consumption figures in the table included the jacket steam. The Rankine-cycle efficiency was close to 70 per cent, and due to developments under way should be increased to 88 per cent. This he thinks will tend not to eliminate the triple-expansion engine.

A very careful study of types of pumping machinery was made when considering the installation of the units. It has been found that if the units are run less than 3500 hours, the centrifugal pump will be the more economical. If they are run more than 3500 hours, the triple-expansion engine will be the more economical. This is without considering the building. As far as the building is concerned, however, the author feels that little difference will be noticed as municipalities are not likely to save money on buildings.

Since this paper was prepared, the city of Milwaukee has purchased another pumping engine of the same size and type as that described in the paper.

[1] Put by Frank O. Wallene, Wallene Engineering Co., Cleveland, Ohio.

TECHNICAL COMMITTEE REPORTS

In this section is given a summary of the codes and standards which have been completed during 1925 and are available either in book or pamphlet form upon application, and of preliminary or tentative drafts which have been printed in *Mechanical Engineering* during the year. Other work of the various committees in process is summarized in the Report of the Council, pages 580-583.

BOILER CODE

Interpretations of the Boiler Code were published in *Mechanical Engineering*, January (1925), p. 56; March, p. 225; May, p. 378; June, p. 515; July, p. 593; September, p. 771; and December, p. 1168. Revisions of portions of Section IV of the Code were given on page 379 of the May issue.

Modifications of the Report of the Sub-Committee of the Boiler Code Committee on the Code for Unfired Pressure Vessels were printed in *Mechanical Engineering*, January (1925), p. 57. This code was issued in complete final form in March, 1925.

Section VII of the Boiler Construction Code, covering Rules for the Care of Power Boilers, was published in *Mechanical Engineering*, May (1925), p. 371.

POWER TEST CODES

Four codes were issued in pamphlet form during 1925, as follows:

Test Code for Condensing Apparatus
Test Code for Displacement Compressors and Blowers
Test Code for Evaporating Apparatus
Test Code for Reciprocating Steam-Driven Displacement Pumps.

Test codes published in *Mechanical Engineering* during the year were:

Test Code for Reciprocating Steam-Driven Displacement Pumps, revision by reorganized committee prior to publication in final form, February, p. 135
Test Code for Centrifugal and Rotary Pumps, tentative draft, March, p. 214
Test Code for Refrigerating Systems, tentative draft, May, p. 365
Test Code for Steam Turbines, tentative draft, September, p. 759
Code on Instruments and Apparatus, preliminary draft of Chapter 21, dealing with condenser-leakage tests, November, p. 920
Test Code on Instruments and Apparatus, preliminary draft of Chapter 3, Temperature Measurement, Section 1—General, December, p. 1161.

RESEARCH

A description of the gear-testing machine designed by Wilfred Lewis for the A.S.M.E. Special Research Committee, was given in *Mechanical Engineering*, January (1925), p. 24, and the February issue, pp. 103-108, contained a report on the progress in steam research, including a report

of the executive committee of the steam table research fund, reports of progress in steam research at Harvard University, the Massachusetts Institute of Technology, and the Bureau of Standards, and a progress report on the Joule-Thomson effect.

SAFETY CODES

The first revision of the Safety Code for Elevators was issued in pamphlet form in July, 1925, and is designated as A17 — 1925. An abstract was published in *Mechanical Engineering*, March (1925), p. 219.

STANDARDIZATION

National (American) Standard Fire-Hose-Coupling Screw Thread: This standard, designated as B26 — 1924, was published in May, 1925. Two supplementary sections were also published entitled, respectively, Production of National (American) Standard Fire-Hose-Coupling Screw Thread, and Field Inspection of National (American) Standard Fire-Hose-Coupling Screw Thread.

Pipe Flanges and Fittings: The February issue of *Mechanical Engineering*, 1925, page 117, contained the first reports of the sectional committee on the Standardization of Pipe Flanges and Fittings. They cover proposed standard dimensions for the following three complete series of fittings:

Malleable-Iron Screwed Fittings designed for 150 lb. Working Steam Pressure

Cast-Iron Screwed Fittings designed for 125 lb. Working Steam Pressure

Cast-Iron Screwed Fittings designed for 250 lb. Working Steam Pressure.

Plain Limit Gages for General Engineering Work: The Tentative American Standard on Tolerances, Allowances, and Gages for Metal Fits, designated as B4a — 1925, was issued in pamphlet form in December, 1925.

Shafting: The Tentative American Standard of the Diameters and Lengths of Shafting, designated as B17a — 1924, was issued in December, 1924, and the Standard for Square and Flat Stock Keys, designated as B17b — 1925, was issued in June, 1925.

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NECROLOGY

ALFRED B. ABBEY

Alfred B. Abbey, of Los Angeles, Cal., died while attending the Pacific Coast Gas Association Convention at Portland, Ore., August 24, 1925. Mr. Abbey was born at Oakland, Cal., September 4, 1884, and was educated in the schools of Guerneville, Cal., at Mt. Tamalpais Military Academy and at Heald's School of Mines. He completed his mining and civil engineering education in France.

His first practical experience was with the Great Eastern Quicksilver Mining Co. at Guerneville, where his father was superintendent. Here he spent three years upon design and construction of apparatus used in mining cinnabar ore, and distilling mercury therefrom. This was followed by a year of similar work with the Sonoma Consolidated Mining Co. Mr. Abbey then went into the oil business as engineer for the Esperanza Oil Co. He was first located at San Francisco and later at Taft, Cal., of which place he served as city engineer for a time. In 1911 he became associated with the Standard Oil Co., spending the next ten years with them in Taft and Whittier, Cal. He began with them on construction work but during the greater part of this period he specialized on the process of extracting gasoline from natural gas by the compression process. After leaving the Standard Oil Co., Mr. Abbey was one of the founders of the supply concern of Neilan, Abbey & Co. of Los Angeles and was vice-president of this concern at the time of his death.

He joined the A.S.M.E. as an associate member in 1917.

STEN ANDERSON

Sten Anderson, engineer and architect of the Tumwater Lumber Mills Co., died at Olympia, Wash., on August 20, 1925. Mr. Anderson, who was born at Ursvoken, Sweden, in 1890, came to this country at the age of sixteen He was an honor student at Cooper Union and was graduated there in 1918 with the degree of B.Sc. in mechanical engineering. He gained his first practical experience as draftsman with Babcock & Wilcox, with the Westinghouse Co., and with the New York Steam Co.

In 1920 Mr. Anderson went to Tumwater, Wash., to develop, at the lumber mill owned by his brothers, the production of fabricated houses. This has grown to be one of the largest industries of its kind in that part of the West.

Mr. Anderson had been a junior member of the Society since 1921.

EDWIN JAMES ARMSTRONG

Edwin James Armstrong, one of the directors of the Erie Steam Shovel Co., Erie, Pa., died on January 21, 1925. Mr. Armstrong was born on May 12, 1861, in Hornby, Steuben County, N. Y. He received his education through home study and began his professional career by

serving an apprenticeship with the Weston Engine Co., Painted Post, N. Y. After two years he was made shop foreman, which position he held for five years. He was subsequently connected with the John T. Noye Manufacturing Co., Buffalo; with the Straight Line Engine Co., Syracuse, N. Y., as shop superintendent; with the Robb Engineering Co., Amherst, Nova Scotia, in the design of engines and shop organization; with the Ames Iron Works, Oswego, N. Y., as designer and general superintendent; and with the Ball Engine Co., Erie, Pa., as chief engineer. When the Erie Steam Shovel Co. took over the latter concern, he became one of its directors.

Mr. Armstrong became a member of the Society in 1886. He belonged to the American Association for the Advancement of Science, to the Paleontological Society of America, to the Erie University Club, and to the Masonic Order.

OLAI BACHE-WIIG

Olai Bache-Wiig, for many years an active engineer in the pulp and paper industry, died at Louisville, Ky., on November 11, 1924. Mr. Bache-Wiig was born at Trygstad, Norway, on June 3, 1876. He received his early education in Norwegian schools, and then went to Germany where he was graduated from the Zwickau Engineering College. After his graduation he returned to Norway and for two years was employed at the Hafslund and the Bohnsdahlen mills.

In 1903 Mr. Bache-Wiig came to the United States where he became an engineer and salesman with Morey & Co. of Boston. After two years with them he spent a year improving the processes of the Mount Tom Sulphite Pulp Co., and a year as consulting engineer for the York Haven Paper Co. of York Haven, Pa., and the Laurentide Paper Co. of Quebec. From 1907 to 1910 he was with the large pulp organization controlled by George Van Dyke. He converted the soda mill at East Angus, Quebec, to the sulphate process and here he produced the first sulphate pulp in America and the first kraft paper made from such domestic pulp. He was engaged in designing a similar sulphate mill to be erected in Vermont when Mr. Van Dyke met his death in an automobile accident, and the project was abandoned.

In 1910 Mr. Bache-Wiig went to Wisconsin where he successfully promoted the Wausau Sulphate Fibre Co. and built their pulp and kraft mill at Mosinee, which he had under production within a year from their inception. It was in this mill in 1919 that he first put into operation his improved kraft machine, which made paper at the then unheard-of speed of 1000 ft. per minute. While with the Wausau concern he also acted as a consulting engineer and as such designed and built large sulphate mills at Ocean Falls, British Columbia, and Bogalusa, La.

In 1923 Mr. Bache-Wiig organized the Tomahawk Kraft Paper Co., which took over a paper mill and power right at Wisconsin Dam. Under his supervision the old mill was remodeled, an addition built, and the power developed. This project was completed in 1924, and at the time of his death Mr. Bache-Wiig was president of the concern. He had been a member of the Society since 1914, and was also a member of the Technical Association of the American Pulp and Paper Industry.

FRANCIS JAMES BADGE

Francis James Badge, tool engineer with the Sperry Gyroscope Co., Brooklyn, N. Y., died on February 25, 1925, of pneumonia. Mr. Badge was born on October 28, 1876, of English parents, at Kobe, Japan. He was educated in the grade schools of Australia and England, and was

graduated from the mechanical course of the Mining School at Camborne, England, in 1897. From 1891 to 1896, he served his apprenticeship as machinist with R. Stephens & Son, Cornwall, England, and then became machinist in the government dockyard of Devonport, later being transferred to the government arsenal at Woolwich.

In 1901, Mr. Badge came to the United States, where he was employed for about a year as a die sinker with the United States Shoe Machinery Co., at Winchester, Mass. For the next nine years, he served in similar capacities with firms in Bridgeport, Conn.; Chicago, Ill.; San Francisco, Cal.; Sydney, Australia; Peterborough, Canada; Horley, England; and Brooklyn, N. Y. From 1910 to 1915, Mr. Badge was machine designer and experimental machinist with the Parkes Machine Co., Brooklyn, N. Y., resigning to become chief draftsman with the Keller Mechanical Engineering Co., Brooklyn. In 1917-18, he was machine designer with Lymburner, Ltd., at Montreal, Canada. From 1918, he was connected with the Sperry Gyroscope Co., and also acted as consulting engineer for the L. S. E. Co., Inc., of Brooklyn.

Mr. Badge had been a member of the Society since 1920.

GEORGE M. BASFORD

George M. Basford, widely known railway publicist and consulting engineer specializing upon the steam locomotive, dropped dead in the Jerome Avenue subway station in New York on Monday evening, October 26, 1925. Mr. Basford was born in Boston on December 27, 1865, attended the public schools there, and in 1889 was graduated from the Massachusetts Institute of Technology with the degree of B.S.

Immediately following his graduation he entered the Charleston shops of the Boston & Maine Railroad where he spent a year as a mechanic. In 1891 he spent some time as a draftsman in the mechanical department of the Chicago, Burlington & Quincy, at Aurora, Ill., and in the motive power department of the Union Pacific. He then spent two years as signal engineer with the Chicago, Milwaukee & St. Paul. During the years 1893 and 1894 he was superintendent of construction with the Johnson Railroad Signal Co., and was signal engineer with the Union Switch & Signal Co. and the Hall Signal Co.

With this wide experience behind him, Mr. Basford began his career as a publicist in 1894, when he became mechanical editor of the *Railway Review*. He left this publication in 1896 to become editor of the *American Engineer and Railroad Journal* (now the *Railway Mechanical Engineer*). This position he held for about ten years. He resigned in September, 1905, to become assistant to the president of the American Locomotive Co., and in March, 1913, he became chief engineer of the railway department of Joseph T. Ryerson & Son. Mr. Basford organized the G. M. Basford Co. in March, 1916, and since then had headed this concern, which specializes in technical advertising. At the same time he was president of the Locomotive Feedwater Heater Co. until it was taken over by the Superheater Co., and at the time of his death was also consulting engineer for the Lima Locomotive Works, Inc.

Mr. Basford had a pronounced influence upon the improvement of steam-locomotive design and utilization and through his keen interest in training methods he helped hundreds of young men to attain high positions in the railway industry. He was also very active in association work, being one of the founders of the Railway Business Association and the Railway Signal Association.

Mr. Basford joined the Society in 1889, and served as Manager from 1906 to 1909, on the Nomination Committee in 1910, on the Publication Committee from 1912 to 1914, and was the A.S.M.E. Representative on the United Engineering Societies Board of Trustees. At the time of his death, he was a member of the Main Committee, Power Test Codes.

ALBERT BENTEL

Albert Bentel, designer and estimator for the American Arch Co. of New York, died on June 19, 1925. He had been a junior member of the Society since 1920.

Mr. Bentel was born in London, England, February 16, 1895. At the age of twelve he came to New York City with his parents. He was graduated from Stuyvesant High School, then took the mechanical engineering course at New York University, from which he was graduated in the year 1918. After six months as tool designer for the Remington Arms Co. of Hoboken, N. J., he became assistant to the production superintendent, working on plant layout and design and supervision of installation of machinery and equipment. After the plant was fully equipped, he coöperated with the foreman to perfect technical processes in connection with small-cartridge manufacture. In 1920, he was designer with the C. H. Mallory Laboratories, Astoria, L. I., and later became sales engineer for the Hyatt Roller Bearing Co. of Newark, N. J. In 1923 he became mechanical and sales engineer for the American Arch Co.

C. EDWIN BERGLUND

C. Edwin Berglund, one of the service engineers of the Worthington Pump & Machinery Co., was killed near Moncton, N. B., on October 5, 1925. The accident happened as Mr. Berglund was riding on a ladder at the side of a locomotive watching the action of a feedwater pump. In passing a freight on a siding he was struck in the head by one of the rear light brackets on the caboose.

Mr. Berglund was born at Spencer, Mass., and received his first engineering education at the Massachusetts Nautical School, 1914-1916. After spending a year as oiler and water tender he became a third assistant engineer of an ocean steamer. Upon the outbreak of the war he passed the second assistant engineer's examination and was commissioned an ensign in the Navy. He served two years and was overseas aboard the U.S.S. *Aroostook*. He was one of those who transported the NC-4 to England prior to its Atlantic flight.

Upon leaving the Navy Mr. Berglund entered Pratt Institute and was graduated in 1923 with the class in industrial mechanical engineering. He joined the Worthington Pump & Machinery Co. in 1923 in their training department, and after three months went on the road for the Locomotive Feedwater Heater Department. Mr. Berglund became an associate-member of the A.S.M.E. in 1923.

WILLIAM J. BEST

William J. Best, New York manager for the Wheeler Condenser & Engineering Co., died at his residence at Westfield, N. J., on September 10, 1925. Born in Philadelphia, January 9, 1864, he received his education in the public schools of that city and at the Franklin Institute. In 1881 Mr. Best entered the works of the Southwark Foundry, in the erecting shop. From there he went into the drafting room, where he completed his apprenticeship in 1885. After leaving the Southwark Foundry, he spent three years at the Midvale Steel Works as foreman, and some time with the Green Fuel Economizer Co. For a number of years following this Mr. Best was connected with the Buckeye Engine Co. He ran the 1000-hour test which this company carried out during the Atlanta Exposition in 1895. He was eastern engineer for the company for five years, manager of the Chicago office for three years, and installed engines and was consulting engineer

upon several large projects, including those of the New Orleans Edison Co., New Orleans Imperial Electric Light Co. of St. Louis, and the St. Joseph Railway & Light Co. Mr. Best had been connected with the Wheeler Condenser & Engineering Co. for more than sixteen years, and was treasurer of the company for a considerable period. He became a member of the A.S.M.E. in 1910.

JOHN B. BIRD

John B. Bird, secretary of Joseph Bancroft & Sons Co., died on February 15, 1925, at Atlantic City, N. J. Mr. Bird had been ill for several months, and had gone to the shore, from his home at Wilmington, Del., hoping to benefit his health.

Mr. Bird was born at Freeport, Pa., in 1870, but moved to Wilmington when he was very young and lived there throughout his life. Following in the footsteps of his father, Mr. Bird studied civil engineering. He served his apprenticeship as recorder on harbor work and in field work on location and construction for the N. & W. R. R. In 1893 he became track supervisor for the B. & O. R. R. and in 1895 went to Joseph Bancroft & Sons Co. There he became chief engineer and in 1912 was elected a director of the company, and, later on, secretary. Some time before his death, he gave up the work of chief engineer, because of his ill health.

Mr. Bird became a member of the Society in 1913.

WILLIAM J. BOHANNON, JR.

William J. Bohannon, Jr., leading draftsman of the valuation department and material inspector for the Seaboard Air Line Railway, Norfolk, Va., died by accidental drowning on March 8, 1925.

Mr. Bohannon was born on October 10, 1893, in Portsmouth, Va. He attended a private school and the Norfolk Academy, then became an apprentice draftsman with the S. A. L. Railway. He was appointed leading draftsman in 1916, and later was leading locomotive draftsman for that company, leaving to serve in the same capacity with the Baltimore & Ohio Railway. During the war, Mr. Bohannon was a private in the United States Marine Corps. Upon receiving his discharge, he returned to the S. A. L. Railway. During the summer of 1924 he took the summer course in shop work at the Massachusetts Institute of Technology He had also completed various technical courses at the Norfolk Mechanics' Institute.

Mr. Bohannon became an associate member of the Society in 1924. He belonged to the American Legion and to the Order of Elks.

WILBUR M. BOSWORTH

Wilbur M. Bosworth, assistant chief engineer of the Standard Steel Tube Co., of Toledo, Ohio, died April 9, 1925. He was born at Baltimore, Md., June 13, 1879, and was graduated from the Baltimore Polytechnic School with the class of 1898.

Until 1911, Mr. Bosworth was with the B.&O.R.R., first as machinist and journeyman, then as junior draftsman, draftsman, and finally leading car draftsman, designing both freight and passenger cars. From 1911 to 1914, he was mechanical engineer with the Kansas City Southern Ry. Co., at Pittsburgh, Kan., and from 1914 on held the same position with the Louisville & Nashville R. R. Co., the Norfolk Southern R. R. Co., the Under-Feed Stoker Co., and the Wine Railway Appliance Co. Some time before his death, he became connected with the Standard Steel Tube Co.

Mr. Bosworth became a member of the Society in 1913.

STEPHEN N. BOURNE

Stephen N. Bourne, who was active in the textile industry for eighty-six years, and who at the time of his death had the distinction of being the oldest bank president in the United States, passed away at his home in Manchester, N. H., on March 25, 1925. Mr. Bourne was born at Attleboro, Mass., on March 12, 1830, and was active up to within the day of his death. On his ninety-fifth birthday he gave out the following recipe for a long and happy life, at the request of The Johns Hopkins University: "Cultivate an even disposition, with absence of worry, a cheerful outlook on the future, optimism in general, an alert mind, and ability to keep away from rheumatism."

Mr. Bourne went to work at Pontiac, R. I., in the textile industry at the age of eight while he was going to school. From 1847 to 1850 he served an apprenticeship as machinist at the Providence Machine Co. on textile machinery. From 1850 to 1867 he was with the Lonsdale Mills, Lonsdale, R. I., where he became master mechanic. In 1867 he went to the Pontiac Mills and Bleachery as superintendent, where he remained until 1881. From 1881 until 1901 Mr. Bourne was agent of the Stark Mills of the Amoskeag Mfg. Co., Manchester, N. H., and from 1901 to the time of his death he held the same position with the Dundee Mills of Hooksett, N. H.

Mr. Bourne was the designer and builder of many textile mills and was the inventor of much improved textile machinery. He was an Honorary Member of the Textile Club of New England and was a 32d degree Mason. He became a member of the Society in 1891.

LEON L. BOWEN

Leon L. Bowen, engineer with Thomas E. Murray, Inc., died suddenly in New York City on Nov. 6, 1925. Mr. Bowen was born at Gallup, New Mexico, on April 12, 1897. After attending school at Las Vegas, New Mexico, and the Santa Monica, Cal., High School he entered Leland Stanford Junior University. From there he transferred to the University of California, and then to the University of Illinois, from which he was graduated in 1922 with a degree in Mechanical Engineering.

Mr. Bowen began his engineering work before completing his college courses, serving as ensign in the U.S.N.R.F. at Mare Island, Cal., during the war. After graduation he went to the Westinghouse Electric & Mfg. Co. at East Pittsburg, where he took the graduate student course. He was promoted to the general engineering department, and then to a similar position at the South Philadelphia works. He joined the New York firm of consulting engineers in 1923.

Mr. Bowen became a junior member of the A.S.M.E. in 1924.

BERT B. BREEDING

Bert B. Breeding, specialist on engineering management problems, died in Long Island City in Dec. 1925.

Mr. Breeding was born on July 24, 1895, at Palouse, Wash. He was graduated in 1917 from the University of Oregon with the degree of B.S. Upon his graduation he entered the U. S. Navy Yard at Bremerton, Wash., as foreman of structural steel layout and assembly, under Commander G. A. Bisset, U.S.N.

In 1919 Mr. Breeding went to Toledo, Ohio, where he spent a year in the training school for junior engineers conducted by H. L. Doherty.

He made a general study of public utility work, including power house, gas, and industrial heating practice.

Upon the completion of this course he spent two years as an apprentice with the Pangborn Corpn. of Hagerstown, Md. At the end of his apprenticeship he became a service engineer for the company and spent two years in the field.

In May, 1925, Mr. Breeding joined the staff of Chas. E. Bedeaux, industrial engineers of New York. He learned the Bedeaux System and worked upon its installation in plants. At the time of his death he was representing Murrie & Co., engineers, on appraisal work with the New York Edison Co.

Mr. Breeding became an associate member of the A.S.M.E. in Nov. 1925.

FRED E. BRIGHT

Fred E. Bright, founder and past-president of The Hess-Bright Mfg. Co., of Philadelphia, died at his home in New York City on Oct. 7, 1925. Mr. Bright was born in October, 1856, at Richland Center, Wis., and attended Mount Union College, Ohio. In 1874 he began to learn the machinist's trade at Cleveland, Ohio, and until 1878 worked in several machine and tool shops in that city.

After 1878 he was engaged in similar lines on his own account, developing many inventions both for himself and for others. Mr. Bright was associate inventor and designer of the linotype system known as the Typograph. In connection with the Ludwig Loewe Co. he established this system in Europe, spending several years in Berlin, Germany, on the building and equipping of a plant for the production of linotype machines. He was later the inventor and designer of a complete linotype. In 1901 Mr. Bright was one of the founders of the Cleveland Cap Screw Co., now the Electric Welding Products Co.

Mr. Bright was one of the founders of The Hess-Bright Mfg. Co. organized in Philadelphia in 1904 for the manufacture of ball bearings. He was vice-president and treasurer of this concern from 1904 to 1912, and was president from 1912 to the time of his retirement not long ago. During the first ten years of this company's existence he developed and patented many inventions relating to ball-bearings and their manufacture. Mr. Bright became a member of the A.S.M.E. in 1913.

OTTO BRINTZENHOFE

Otto Brintzenhofe, assistant chief engineer of the Electric Vacuum Cleaner Co., Cleveland, Ohio, died on April 10, 1925, at his home in Willoughby after a long illness.

Mr. Brintzenhofe was born at Fort Wayne, Ind., on April 29, 1887. He was educated in public schools of that city and by evening study in mechanical engineering while serving an apprenticeship in the machine shop of the Western Gas Construction Co., which he began at the age of seventeen and continued for three years. He then obtained a position as draftsman at the Fort Wayne Works of the General Electric Co. He was with this concern for several years, eventually becoming a designer. Mr. Brintzenhofe also worked for a time for the Wayne Tank & Pump Co. and for the Remy Electric Co. in experimental and designing work. Through his experience upon small electric motors he became associated with the Electric Vacuum Cleaner Co. in 1918.

Mr. Brintzenhofe was a member of the Masonic Order. He became an associate member of the Society in 1923.

CHARLES W. BURROWS

Charles W. Burrows, research engineer in magnetism, died at South Orange, N. J., on May 2, 1925. Mr. Burrows, who was born in Ontario, Canada, on July 2, 1874, received his early education in the schools of Detroit. He then attended the University of Michigan, where he specialized in science. He was graduated in 1898 with the degree of A.B. and was awarded his degree of M.A. in 1901 and Ph.D. in 1907. After graduation he was instructor in science at the Detroit High School until 1905, and spent the following year as a graduate student and instructor in physics at the University of Michigan. He went to the Bureau of Standards in 1906 and remained there until 1918, being successively laboratory assistant, assistant physicist, and associate physicist, and for seven years in charge of the Magnetic Laboratory. During the war he was closely associated with the Naval Consulting Board and the Navy Department.

Dr. Burrows began his consulting engineering work in 1910, and since 1918 had given his entire time to magnetic and metallurgical investigations for large corporations. He was the inventor of a magnetic permeameter, a magnetic deflectoscope, a magnetic analyzer, and several other complicated devices, and for several years prior to his death was vice-president of the Burrows Magnetic Equipment Corpn. and chief engineer of the Burrows Train Control Co. Dr. Burrows was a prolific writer and a great many of his valuable papers upon magnetic subjects are incorporated in the proceedings of the technical societies and in the publications of the Bureau of Standards. He was a Fellow of the American Institute of Electrical Engineers, and was a member of the Engineers' Club of New York, the American Society for Testing Materials, the American Society for Steel Treating, and of the A.S.M.E., which he joined in 1919.

FREDERICK B. CAPEN

Frederick B. Capen, superintendent of the Caspar Works of the Texas Co., Caspar, Wyo., died on January 2, 1925. Mr. Capen was born on March 25, 1891, in New York, N. Y. He was graduated from Yale University in 1912 and then entered the service of the Texas Co. at their Port Arthur Works. The following year he was promoted to the position of mechanical engineer of the works and two years later became fuel engineer for the refining department. In 1917 Mr. Capen was appointed efficiency engineer with headquarters at Port Arthur, Texas, which position he held until his appointment as superintendent of the Caspar Works in June, 1922. It was under his supervision that the Caspar refinery was built. Mr. Capen became a junior member of the Society in 1917.

EDWARD WHEELER CARR, JR.

Edward Wheeler Carr, Jr., president of E. W. Carr, Inc., of New Orleans, La., died on February 22, 1925. Mr. Carr was born in Brooklyn, N. Y., in 1879. He studied mechanical engineering at Cornell University. Beginning work in the early part of 1902 as designer of a line of physical and physiological laboratory apparatus for the Chicago Laboratory Supply & Scale Co., he continued as draftsman with D. H. Burnham & Co., architects, Sargent & Lundy, engineers, and with Thomas & Smith, Chicago, until January, 1904. He then went to Montreal, P. Q., as assistant engineer to superintend the installation of heating and ventilating work in the Hochelaga shops of the Canadian Pacific R.R. He continued with the C.P.R. until May, 1905,

and thereafter was estimator and superintendent with Thomas & Smith, manager of the engineering and specialty department of The Ahrens & Ott Mfg. Co., and designing engineer and superintendent of construction for E. A. Sammons Co., Ltd., all of New Orleans, La.

In July, 1912, Mr. Carr became a partner in the firm of Carr & Scott, heating and ventilating contractors of New Orleans, and in May, 1914, established the business of E. W. Carr, Inc., specializing in mechanical equipment for buildings, insurance appraisal of machinery losses, and preparation of plans and specifications for mechanical equipment for buildings.

Mr. Carr was an associate of the Society from 1916 to 1921, and in the latter year became a member. From 1918 to 1920, he was secretary of the New Orleans Local Section, and in 1920-21 was chairman of the Section.

WALTER CASTANEDO

Walter Castanedo, merchant engineer of New Orleans, La., died on January 11, 1925. Mr. Castanedo was born in August, 1873, in New Orleans. He attended the University of Notre Dame for two years and then for three years pursued special engineering studies at Tulane University. He installed the first turbine-driven centrifugal pumps in the Atlanta water works and designed the Jefferson and Plaquemines drainage district. He was the New Orleans representative of large power-plant manufacturers, including the DeLaval Steam Turbine Co., the Cochrane Corpn., the Wheeler Condenser & Engineering Co., etc. For the past twenty-five years he was in business for himself in New Orleans as a merchant engineer specializing in power-plant equipment. Mr. Castanedo became a member of the Society in 1907.

JOHN DOUGLAS CHERRY

John Douglas Cherry, president of the Roto Co., Hartford, Conn., died on January 3, 1925. Mr. Cherry was born on January 2, 1875, in Pittsburgh, Pa. He attended the Western University of Pennsylvania, now the University of Pittsburgh. For six years he was connected with the French Works of the Railway Steel Spring Co., as assistant to the manager. From 1906 to 1908 he was assistant to the manager of the air-spring department of the Crucible Steel Co., resigning to do independent sales work as manufacturers' agent, specializing in power-plant equipment. In 1917 he became associated with the Roto Co. as general sales manager and the following year was elected president, which position he held at the time of his death. Mr. Cherry became an associate of the Society in 1917. He belonged also to the American Society of Civil Engineers and to the Hartford Engineers' Club.

WILLIAM W. CHRISTIE

William W. Christie, consulting engineer and architect, Ridgewood, N. J., died on April 13, 1925. Mr. Christie was born on July 12, 1866, in Paterson, N. J. He was graduated from the Paterson High School in 1882 and then spent two and one-half years in the drawing room of the Rogers Locomotive Works. Later he became engineer of the Ramapo Iron Works in charge of the foundry and car departments. Mr. Christie was engaged in consulting and architectural work for many years in Paterson, N. J., and transferred his business interests to Ridgewood in 1918. He became a member of the Society in 1889.

HERBERT HAMILTON CLARK

Herbert Hamilton Clark, manager of the Los Angeles office of Chas. C. Moore & Co., engineers, died at Los Angeles, Cal., on June 5, 1925. Mr. Clark was born in England in 1860, and received his early education at Queen's Road Academy, the Northern Institute, and Rugby. At the age of seventeen, he became apprentice at Jacks Foundry, where he remained for three years. Following some experience in large shops in England, he went to sea, and worked his way up from oiler to chief engineer on the steamship *City of Berlin*, running between New York and Liverpool.

Leaving the sea in 1886, Mr. Clark worked for the firm of B. F. Kelly & Son, of New York, selling the Berryman heater. In 1891 he joined the Fishkill Engine Co. as engineer and salesman. In 1896, Mr. Clark went to San Francisco, where he engaged in machinery sales work on his own account for two years. In 1898, he joined Chas. C. Moore & Co. of San Francisco as engineer and salesman. For three years he travelled along the coast and to the Hawaiian Islands, spending a large part of two years in Honolulu. In 1901, the company sent him to Los Angeles as the manager of the Los Angeles office of Chas. C. Moore & Co. and the Babcock & Wilcox Co. In this capacity, he was identified with the power-plant development of the Pacific Coast until the time of his death.

Mr. Clark became a member of the Society in 1908.

SAMUEL J. CLARKE

Samuel J. Clarke, naval engineer and constructor, died at his home in New York, N. Y., on December 30, 1924. Mr. Clarke was born at Boston, Mass., on August 19, 1849. After attending the Boston schools, he entered Columbia University, where he studied mechanical engineering. After completing his college work, Mr. Clarke entered the Morgan Iron Works in New York as an apprentice, serving as such from 1864 to 1868. After this he served as assistant engineer on various steamers sailing from New York. In 1874 he was sent to Japan where he remained for over ten years. He served as chief engineer on the steamship *Kunda*, and several others in Chinese and Japanese waters, and was superintendent of construction of the steamship *Kamtchatka*.

After spending a considerable period at Renfrew, Scotland, in the shipbuilding trade, he went to Paris as the representative of a group of American capitalists interested in shipping matters. Upon his return to the United States, Mr. Clarke became connected with the Atlantic Mutual and remained with this company until he retired several years ago.

Mr. Clarke, who was also a member of long standing in the American Society of Naval Architects and Marine Engineers, joined the A.S.M.E. in 1884.

CHARLES C. COBB, JR.

Charles C. Cobb, power engineer for the Walworth Company at Kewanee, Ill., died of pneumonia at his home in that city on Nov. 17, 1925. He was born at Chicago, Ill., on May 26, 1889, and received his early education at the Hyde Park school. After graduation there he entered the University of Chicago, where he remained three years studying mechanical engineering. He then transferred to the University of Illinois, where he completed his work, being graduated in 1911 with the degree of B. S.

Mr. Cobb started his work as engineer on comparative tests of saturated and supersaturated steam on locomotives of the Illinois

Central Railroad. After a few months he joined the National Tube Co. at Kewanee, Ill. He began work in their engineering department on power-plant and machine design, then later was detailed to special power investigations. In 1918 he was promoted to the position of general foreman of their main power plant. He became connected with the Walworth Company in 1920.

Mr. Cobb was a member of the Masonic order, and was very active in civic affairs in Kewanee. He joined the A.S.M.E. in 1912.

OLAV S. CORNELIUSSEN

Olav S. Corneliussen, engineer of portland cement plants, died suddenly in Sao Paulo, Brazil, in September, 1925. Mr. Corneliussen was born at Christiania, Norway, on Apr. 28, 1886. In 1906 he was graduated with the degree of M. E. from Skiensfjordens Mekaniske Fagskole at Porsgrund, Norway.

After graduation he worked for a few months as a draftsman at Tuhnes Mekaniske Varkated at Christiania, then went to New York where he became a draftsman for McClintic-Marshall Co. He left to join the F. L. Smidth & Co. of New York, and for four years he was engaged with them in engineering work upon cement plants and cement machinery.

With the experience gained there, Mr. Corneliussen became a specialist in cement-plant engineering and for the next ten years carried out important work upon mills and machinery for the Kent Mill Co., New York, the Whitehall Portland Cement Co., Cementon, Pa., Geo. S. Emerick and the Phoenix Portland Cement Co., Nazareth, Pa., the Clinchfield Portland Cement Co., Kingsport, Tenn., the Kentucky Portland Cement & Coal Co., Louisville, Ky., the Atlas Portland Cement Co., Northampton, Pa., and with Albert C. Wood, construction engineer in Philadelphia.

In 1917 Mr. Corneliussen went to Cuba, where he spent two years as mechanical engineer of the Cuban Portland Cement Co., a year as consulting engineer, and also a year as director and mine prospector for the Compania Campo de Hierro at Isla de Pino. In 1922 he returned to Nazareth, Pa., to become mechanical engineer for the Dexter Portland Cement Co. in the remodeling and enlarging of their plant. About a year ago he went to Sao Paulo, Brazil, where at the time of his death he was associated with the Foundation Co. and Montreal interests in the construction of a large cement plant.

Mr. Corneliussen joined the A.S.M.E. in 1915.

CHARLES H. COVE

Charles H. Cove, master mechanic of the West Warren Cotton Mills, died at Springfield, Mass., Sept. 26, 1925. Mr. Cove was born in Somersworth, N. H., April 13, 1892. He was educated in the public schools there and also took two courses in the International Correspondence School. His early apprenticeship was served in the Massachusetts Cotton Mills of Lowell, Mass., with whom he remained until 1917. He spent several years as machinist engaged in the operation of general steam engines including high-duty steam turbines and served an apprenticeship as draftsman in steam-engineering and boiler-room work. From 1914 to 1917, as draftsman for the concern, he designed and laid out their larte dye house proposition in Lindale, Ga., and erected new power plants in Lowell.

Mr. Cove became connected with the Warren Cotton Mills in 1917, and as master mechanic and engineer for that concern rearranged the machinery in their new mills, replacing the return tubular boilers with

water-tube units and substituting oil for coal fuel. From 1919 until the time of his death he had complete charge of all mechanical and construction work for the plant.

Mr. Cove became a member of the A.S.M.E. in 1919 and was also a member of various other technical and fraternal organizations.

FRANK CREELMAN

Frank Creelman died suddenly of heart trouble on January 19, 1925, at Schenectady, N. Y. He was associated with the Switchboard Department of the General Electric Co.

Mr. Creelman was born in Montreal, February, 1864, and received his early education in the public schools of that city. He was graduated from Cooper Union in 1887 with the M.E. degree, and served his machinist's apprenticeship with John Robertson & Co. of Montreal, and the Delamater and Morgan Iron Works in New York. From 1884 to 1888, Mr. Creelman was superintendent of works with the Marvin Safe Co., then took charge of design and construction with the Dynamite Gun Co., as chief assistant engineer, and supervised the acceptance trials at the Government Proving Grounds, Sandy Hook. About this time, a revolution occurred in Brazil, and the Brazilian government purchased from the United States the gunboat *Vesurius*, which had been equipped with one of the dynamite guns. Mr. Creelman accompanied the *Vesurius* to South America as expert engineer for the purpose of instructing the officers and crew in the handling of the dynamite gun, but the revolution had ended before the arrival of the *Vesuvius*.

In 1892, Mr. Creelman became chief engineer of the American Electric Elevator Co., and later was associated with Otis Brothers & Co. (now the Otis Elevator Co.) after the merging of these companies. In 1895, he accepted a position as chief engineer for Thomas L. Wilson and his various Canadian companies, for the manufacture and utilization of calcium carbide, during which time he designed and superintended the construction of a number of hydroelectric power plants, and special electric furnaces and machinery used in the manufacture of calcium carbide. He designed and superintended the manufacture of the gas-lighted and whistling bell buoys, now so universally used in this and other countries.

From 1908 to 1910, Mr. Creelman was managing engineer at the Smyser Machine Works (Arbuckle Brothers), and from 1910 to 1911 was chief engineer for the Union Bag and Paper Co., Canada. In 1912, he returned to this country and became chief engineer of the Gas and Oil Combustion Co. in the development of "surface combustion." From 1914 to 1918, he was with the Good Inventions Co., assisting in the development of inventions chiefly relating to internal-combustion engines and the use of kerosene in automobile engines.

During the war, Mr. Creelman was at the U. S. Naval Experimental Station, New London, working on the design of anti-submarine apparatus. After the war, he did special navy work with the General Electric Co. Mr. Creelman was a member of the Associated Veterans of the Delamater Iron Works and at the time of his death was the president of the Association. He became a member of the Society in 1894.

RALPH C. DAVISON

Ralph C. Davison, salesman and consulting engineer for the American Abrasive Metals Co., died on April 15, 1925. Mr. Davison was born at Hoboken, N. J., in 1875, and was educated at the German Academy and at Stevens Preparatory School. His first work was as tracer and draftsman for the Scott Printing Press Works and the

Worthington Pump Works, and then from 1895 to 1898 he served as special apprentice in the shops of the Central R.R. of N.J. at Elizabethport. From 1898 to 1902, he worked as draftsman and designer, and during the following seven years was associate editor of the *Railroad Gazette*. For two years, he had charge of the drop-forge shop of Streeby & Foote; from 1911-12 was secretary of the Concrete Association of America; and from 1912 to 1918 was engaged in developing improved methods of manufacture and selling for The American Macon Safety Tread Co. He became connected with the American Abrasive Metals Co. in 1918.

Mr. Davison was the author of a book on Concrete Pottery and Garden Furniture. He was a member of the American Society of Safety Engineers, as well as of the A.S.M.E., which he joined in 1921.

ARTHUR MALCOLM DEAN

Arthur Malcolm Dean, chief engineer of the General Chemical Co., died on July 4, 1925, of septic poisoning. Mr. Dean was born at Pembroke, Mass., on October 25, 1883, and attended school in that town and at Canton, Mass. He studied mechanical engineering at the Massachusetts Institute of Technology and was graduated in 1905 with the degree of B.S. He entered the then new profession of automotive engineering with the Pope Mfg. Co. at Hagerstown, Md. At the time this plant was closed in 1907 he was the assistant superintendent.

From 1907 to 1909 Mr. Dean was assistant mechanical engineer of the Mora Co. in Newark, N. Y., and for the next four years chief engineer of the Mathison Automobile Co. at Wilkes-Barre, Pa. During this last period he was also consulting engineer with the Palmer-Singer Mfg. Co. of Long Island City, N. Y. In 1914 Mr. Dean went to Cleveland as assistant engineer of the Ferro Machine & Foundry Co. Upon the formation of the Templar Motor Corpn. in that city he was chosen as vice-president and chief engineer, being the designer of the Templar car. At the same time he was consulting engineer with the Rubay Body Co. and the Swan Carburetor Co. He became chief engineer of the General Chemical Co. in May, 1924, and was active in this work in New York City up to within a few days of his death.

In addition to his work upon motor cars Mr. Dean did much work upon standard marine motors and those of the Diesel type. Mr. Dean joined the Society as a junior in 1907 and had been a full member since 1913. He was also an active member of the Society of Automotive Engineers.

EDMUND F. DEINERT

Edmund F. Deinert, superintendent of the pumping stations of the Hawaiian Commercial & Sugar Co. of Puunene, Island of Maui, met his death in the power plant of his company on Oct. 26, 1925. Mr. Deinert was pulling a knife switch in the transformer tower, when a short circuit sent 22,000 volts through his body, causing instant death.

Mr. Deinert was born in Germany in January, 1876. Early in life he came to California and his education was received in the public schools there and in the Hawaiian Islands. He began his mechanical work as apprentice in a locksmith's shop in San Francisco in 1895. After eight months he transferred to a machine repair shop, where he remained two years. In 1897 he went to Honolulu and worked for a time in a small repair shop. From 1898 until 1901 h served a more complete apprenticeship, this time at the Honolulu Iron Works.

Mr. Deinert undertook his first large sugar machinery job when he erected equipment for a new mill of the McBryde Sugar Co., Ltd., on

the Island of Kauai, in 1901. That same year he undertook the erection of a mill for the Hawaiian Commercial & Sugar Co., at Puunene, and upon its completion remained with them as foreman of their machine shop. He was appointed chief engineer in 1907 and was responsible for the installation of a very complete system of pumping stations, requiring some 5000 hp. At the time of his death he was just opening their new power plant at Kaheka.

Mr. Deinert became a member of the A.S.M.E. in 1914.

CARL GEORGE DELAVAL

Carl George deLaval, specialist upon centrifugal pumps and turbine machinery, died on October 5, 1925. He was born on April 16, 1862, at Gothenburg, Sweden, and was educated in that city at the College of Gothenburg and at the Chalmers Institute. In 1880 he was given the degree of M.A. in civil and mechanical engineering. After serving a two-year apprenticeship in the machinery department of the Gothenburg Ship Yards, Mr. deLaval came to America, where in November, 1882, he entered the employ of the Knowles Steam Pump Works at Warren, Mass., as a draftsman. His promotion was rapid and he rose to be chief engineer, superintendent, and eventually manager. After the consolidation of the concern with The Geo. F. Blake Mfg. Co., Mr. deLaval was made general manager of the resulting Blake-Knowles Steam Pump Works at East Cambridge, Mass. This position he held until 1904, when he joined the Henry R. Worthington organization as manager and a director. In 1915 he resigned to become connected with T. A. Gillespie, and was made vice-president and general manager of the American Shell Co., the East Jersey Pipe Corpn., and the International Steel & Ordnance Corpn. He remained with these concerns until 1921. Since that date he was with the Machinery Engineering Sales Co. and the Blair Securities Co. in New York City.

Among Mr. deLaval's achievements may be mentioned the design, erection, and equipping of the plant of the Knowles Co. in 1887, the designing and patenting of more than twenty-five improved machines for Blake-Knowles, the organization of the works and development of a line of products for Worthington, and active work in bringing about the merger which resulted in the formation of the Worthington Pump & Machinery Co. He was also a pioneer in the installation of electric drives for pumps and sugar machinery and in the application of the Edison three-wire D. C. system for the full operation of industrial plants. He wrote several books on centrifugal and turbine pumps and was the holder of forty-four patents in this field.

Mr. deLaval was a member of the Society of Naval Architects and Engineers, the Society of Naval Engineers, the Society of Swedish Engineers, and the John Ericsson Memorial Society of Swedish Engineers. He had been a member of the Society since 1886.

AUSTIN R. DODGE

Austin R. Dodge, consulting engineer with the General Electric Co. at Schenectady, N. Y., died there on Aug. 29, 1925. Mr. Dodge was born at Southbridge, Mass., in 1869, and attended school at Phillips Academy at Andover, Mass. He later entered the Worcester Polytechnic Institute, from which he was graduated in 1893 with the degree of B. S. in electrical engineering. This was followed a year later by his master's degree.

Mr. Dodge went to Schenectady thirty-one years ago and spent a year with C. G. Curtiss upon the development of his multi-stage impulse turbine. This was the forerunner of the turbines now built by the General Electric Co., with which concern Mr. Dodge became associated

at the end of his year of experimental work. During his years with this company he specialized upon turbines, superheated steam, and fluid meters. He was the patentee of several important inventions and was awarded an M. E. degree for his work some years ago.

He joined the A.S.M.E. in 1907.

FREDERICK H. DODGE

Frederick H. Dodge, for many years one of the prominent mechanical engineers in the city of Toledo, Ohio, died suddenly on Aug. 12, 1925, in Maine while on a vacation in the Rangely Lake region. Mr. Dodge was born at Newburyport, Mass., on Dec. 26, 1866 and went to Toledo with his parents two years later. He received his education in the public schools there, and at the Massachusetts Institute of Technology, from which he was graduated with the degree of B. S. in 1890.

He began his active engineering career immediately after graduation with the Bissell, Dodge & Erner Co., manufacturers of electrical equipment at Toledo. After being with them six years he became an official of the Toledo White Lime Co.

Mr. Dodge was one of the pioneer automobile manufacturers in Toledo. From 1910 to 1914 he was connected with the Ohio Electric Car Co. and later was treasurer of the Milburn Wagon Works during their motor manufacturing days. Of late he was identified with many enterprises, including that of the Secor Hotel Co. Mr. Dodge joined the A.S.M.E. in 1914.

CHARLES F. DRAKE

Charles F. Drake, engineer with the Babcock & Wilcox Co. in Chicago, died in that city on November 12, 1925. Mr. Drake was born on February 9, 1875, at Chicago. He was educated in engineering at the University of Michigan, which he attended between 1892 and 1896. While attending the University he spent his summer vacations as an apprentice machinist with the C. R. I. & P. R. R. In 1896 he went to work for Swift & Co. in their testing laboratory at Chicago, on coal analysis, flue-gas analysis, etc. In 1897 and 1898 he was mechanical engineer with the Northwest Copper Co. at Homestead, Ore., doing much work on mine ventilation. He became engineer of tests with the Chicago, Rock Island & Pacific Railway in 1898. He designed and built their testing laboratories, and installed a 300,000-lb. Riehle testing machine therein. Mr. Drake became connected with Sargent & Co., steel founders, in 1901, as mechanical engineer and laid out a number of departments for them. After this he spent a year with Robert W. Hunt & Co. of Chicago as engineer in charge of the construction of the Egyptian Portland Cement Works at Fenton, Mich. Mr. Drake, in 1904, joined the Stirling Boiler Co. of Chicago. In 1907 this became a part of the Babcock & Wilcox Co., with which Mr. Drake remained up to the time of his death. He specialized in power-plant installation with this company. Mr. Drake became a member of the Society in 1914.

GEORGE F. DRAKE

George F. Drake, inventor of lock-nuts and other hardware specialties, died at San Francisco on September 12, 1925. Mr. Drake, who was born in San Francisco on November 3, 1869, left school at the age of fourteen and was apprenticed to a watchmaker. During his apprenticeship he constructed several rather intricate working models, including a steam engine of the "donkey" type. He continued as a

watchmaker until 1891, when he turned to merchandise selling, spending the next twenty-one years largely on the road. In 1912 he became a contractor, spending two years on building construction and two years more upon mine equipment.

It was during his mine experience that he conceived his double lock-nut which now bears his name. In 1917 he established a plant at San Francisco for their quantity manufacture, this being the first extensive automatic screw machine installation on the Pacific Coast. When some three years later the demand outgrew the capacity of the plant, Mr. Drake associated his business with that of the National Screw & Mfg. Co. of Cleveland, Ohio. In June, 1925, this Cleveland concern took over the entire business of the Drake Lock-Nut Co. and Mr. Drake turned his attention to the Drake Co. in San Francisco, manufacturing a furniture glider. Among his other activities in the engineering field outside of his regular line, Mr. Drake during the war developed a successful traction engine. He became an associate-member of the Society in 1919.

GEORGE P. DRAVO

George P. Dravo, president of the Dravo Co., Milwaukee, Wis., died on May 11, 1925, after a long illness. Mr. Dravo was born on February 13, 1868, in Pittsburgh, Pa. He was graduated from Lehigh University in 1888 as mechanical engineer and then took one year of graduate work. He served his apprenticeship with Fraser & Chalmers, Chicago, and in 1895 became connected with the Nordberg Manufacturing Co. in Milwaukee and was with that firm for six years. He resigned to open engineering and contracting offices. In 1916, he established the Dravo Co. to manufacture steam superheaters. Mr. Dravo became a junior member of the Society in 1889 and a member in 1906. He was the first president of the Milwaukee Engineering Society and was a member of the Psi Upsilon fraternity.

LYNN B. EASTON

Lynn B. Easton, manager of the Laidlaw Works, Cincinnati, Ohio, of the Worthington Pump & Machinery Corpn., died on January 24, 1925. Mr. Easton was born in Newport, Ky., on January 26, 1885, where he attended grammar and high school. In 1903 Mr. Easton entered the employ of the Laidlaw Co. as timekeeper, and later occupied positions as payroll and cost clerk. In 1913 his initiative and knowledge of the product gained him a position with the St. Louis sales office and in 1919 he was appointed manager of that office. Early in 1920 Mr. Easton returned to the Laidlaw Works as production superintendent and three years later was appointed works manager.

Mr. Easton became an associate of the Society in 1916. He was actively interested in civic affairs and was a member of the Cincinnati Chamber of Commerce. He was a member of the National Metal Trades Association and of the National Foundrymen's Association. He belonged to the Masonic Order, Shriners, and Knights of Pythias.

CARL J. EHBETS

Carl J. Ehbets, veteran official of Colt's Patent Fire Arms Mfg. Co., and one of the world's leading firearms patent experts, died at Hartford, Conn., on July 18, 1925. Mr. Ehbets was born in Germany in May, 1845, and received his education in that country. In 1861 he began a machinist's apprenticeship and at the same time carried on

his technical studies at the Jessen Polytechnic Institute in Hamburg. From 1864 to 1867 he served an apprenticeship as gunsmith at the famous Krupp Works and also studied mechanical engineering at Karlsruhe.

Mr. Ehbets came to the United States in 1868 and joined Colt's Patent Fire Arms Co. as mechanical engineer upon the Gatling gun. From 1883 up to the time of his death he gave his attention almost entirely to patent work for the company, and he assembled at the Colt armory one of the most complete firearms patent libraries in existence. He handled all the patents of John M. Browning, including those on the well-known machine gun and automatic rifle produced by this inventor at the time of the World War. Mr. Ehbets spoke and wrote six languages and so was able to conduct first-hand patent investigations in foreign countries and to carry on international correspondence relative to firearms development.

Mr. Ehbets became a member of the Society in 1890.

CAPT. WILLIAM E. ELLIOTT

Captain William E. Elliott, for years one of the best known marine men on the Great Lakes, died of the infirmities of old age at his home in Manitowoc, Wis., on Dec. 28, 1925. Captain Elliott was eighty-five years of age, having been born at Nottinghamshire, England, on Sept. 14, 1840. He came to America with his parents in 1854, and located at London, Canada, where he obtained his schooling. He began his engineering work at the age of seventeen with the Great Western Railway, with whom he remained in a mechanical capacity for six years. This was followed by two years with the Detroit Locomotive Works.

In July, 1863, Captain Elliott embarked upon what was to be his real life's work, when he secured a berth as assistant engineer of the steamship *Reindeer* on the Saginaw River. He was appointed chief engineer of the Ward Line in 1865, and shortly after went with the Goodrich Transportation Co., as chief engineer of their steamship *Alpena*.

Captain Elliott was promoted to be chief engineer of this company in 1897 and held this position until his retirement three years ago. Since that time he still served the company as consulting engineer and was frequently called upon for advice. As chief engineer Captain Elliott directed the building of many large lake steamers and placed the machinery in many others. He was the inventor of many improvements in marine practice which are today applied upon ocean steamboats as well as upon the lakes.

Captain Elliott was a very popular man and was a member of numerous fraternal, civic, and engineering organizations. He served as national vice-president of the Marine Engineers' Beneficial Association, and had been an active member of the A.S.M.E. since 1889.

HALCOLM ELLIS

Halcom Ellis, adding machine-inventor, died on May 26, 1925, at Sévres, France. Mr. Ellis was born at Springfield, Mo., on October 17, 1867, and at the age of fourteen he entered the first manual training school ever established in St. Louis. His later training consisted of a special course in mining engineering, two years in architectural drafting, a general course at Washington University, and a law course at the St. Louis Law School. He was given the degree of LL.B. by Washington University in 1891.

Mr. Ellis began his practical engineering work as a teacher of mechanical drawing at the St. Louis Manual Training School in 1889,

In 1891 he began the practice of patent law in St. Louis, at the same time designing and superintending the building of devices for his clients. In 1899 he invented the Ellis Adding Typewriter. Shortly afterward its manufacture was established at Newark, N. J., and Mr. Ellis had full charge of this concern until his retirement in 1919, after which time he lived abroad. He became a member of the Society in 1911 and was also a member and a medalist of the Franklin Institute.

JOHN A. ENOS

John A. Enos died at Torrence, Cal., on August 31, 1925. He was born at St. Louis, Mo., on April 11, 1880, and left school at the age of sixteen to become an apprentice in the machine shops of the Missouri Pacific Railroad. By correspondence study he trained himself in mechanical drawing and was promoted to the drafting department of the shops, where he remained until 1902. He then spent a year each with the St. Louis Plate Glass Co. as erecting machinist, with the St. B. & T. R. R., in the same capacity, and with the Moon Motor Car Co. as foreman. In 1905 he moved to Ardmore, Pa., and spent two years as foreman with The Autocar Co. After this he returned to St. Louis, where he became chief inspector of the automobile department of the St. Louis Car Co., and later superintendent of the A. B. Cole Motor Car Co. From 1911 to 1913 Mr. Enos was supervisor of tools for the Pierce Arrow Motor Car Co.; 1913 to 1916 representative in the Middle West of The L. S. Starrett Co., of Athol, Mass.; 1916 to 1917 works manager of the Jones Motor Car Co., of Wichita, Kan.; 1917 to 1918 vice-president and works manager, Homer Laughlin Engineering Corpn. of Los Angeles; and 1918 to 1920 assistant to the vice-president of The Buda Co., Harvey, Ill. Since 1920 Mr. Enos was connected with the Western Forge & Mfg. Co. of Los Angeles, the Union Tool Co. of Torrence, Cal., and the General Petroleum Co. of Los Angeles. He was a 32d degree Mason. Mr. Enos joined the Society in 1920.

WILLIAM EDGAR FIRTH

William Edgar Firth, retired, died January 4, 1925, at his home at Wilmette, Ill. Mr. Firth was born at Sheffield, England, June 30, 1856, and received his education at the Regent Lane Academy, Sheffield, and the South Kensington Science and Art Museum, London. Mr. Firth was successively connected with Vickers, Maxim & Co., Chas. Camel & Co., and Brown, Bailey & Dixon, all of Sheffield, and later with John Fowler & Co., of Leeds, England. After arriving in the United States, he became associated with the Midvale Steel Co. at Nicetown. He remained with that company for thirty-five years, succeeding John H. Taylor as chief engineer and acting in that capacity until the inauguration of the safety-first movement, when he became chief safety engineer. He organized the Safety Department for Midvale and continued the safety work, and also acted in a consulting capacity, until he retired from active business in 1917. He served in the government ordnance department during the war.

Mr. Firth became a member of the Society in 1909, and was also a member of the Iron and Steel Institute and of the American Institute of Electrical Engineers.

ELBERT C. FISHER

Elbert C. Fisher, vice-president and general manager of the Wickes Boiler Co., Saginaw, Mich., and well known as an engineer and designer of steam boilers, died suddenly on May 18, 1925. Mr. Fisher

was born on January 1, 1865, in Scranton, Pa. He prepared for college in the School of Lackawanna, Scranton, and then entered Cornell University, from which he was graduated as a mechanical engineer in the class of 1890.

For one year he worked in the locomotive repair shop of the Delaware, Lackawanna & Western Railroad in Scranton, and then went for a year to Westinghouse, Church, Kerr & Co., in Chicago. For four years he was manager of the Chicago office of the Murphy Iron Works, resigning to become associated with Wickes Bros. in Saginaw, Mich., taking charge of the boiler department and holding that position until 1907 when the present Wickes Boiler Co. was organized and he became its vice-president and general manager.

Mr. Fisher was regarded as one of the country's leading experts in the design and operation of steam boilers. He was long a member of the American Boiler Manufacturers' Association and was honored by election to its vice-presidency. Two years ago he was urged by that organization to accept its presidency, but declined to do so. He became a member of The American Society of Mechanical Engineers in 1891 and attained prominence on the A.S.M.E. Boiler Code Committee, taking a most important part in the difficult work of formulating the first edition of the Boiler Code in 1914. Mr. Fisher served faithfully as member of several of the important sub-committees of the Boiler Code Committee, particularly those involving mathematical research, and he was active in this committee work up to the end. He was one of the managers of the Society from 1919 to 1922 and a member of the A.S.M.E. Council for the same period. He was also an executive of the Board of Boiler Rules for the State of Michigan.

Through his studies in boiler design Mr. Fisher became interested in the problem of water supply, and for several years gave close study to the subject. He was one of the leading advocates of the use of Saginaw Bay water for that city's water supply, and spent an immense amount of time making studies of the bay and other water sources to support his opinion.

ANDREW FLETCHER

Andrew Fletcher, president of the American Locomotive Co., died on Nov. 29, 1925, at his home in New York. He was born in New York on June 8, 1864, and received his education at the College of the City of New York, followed by courses in naval architecture and marine engineering.

Mr. Fletcher's interests at first were centered upon manufacturing, but of late years his activities extended beyond the metal industries into the fields of oil and banking.

His list of directorships, included, besides the great locomotive company of which he was the head, such concerns as the American Car & Foundry Co., the Consolidated Iron Works, North River Iron Works, the Montreal Locomotive Works, the Superheater Company, William Cramp & Sons Ship & Engine Building Co. and the Bucyrus Company. Outside of these metal industries his directorship of such institutions as the Chase National Bank, the Atlantic Gulf Oil Corpn. and the Gulf & West Indies Steamship Lines were representative.

Mr. Fletcher belonged to many well-known organizations, including the Engineers' and the Railroad Clubs, the Institution of Engineers & Shipbuilders (Scotland), and the American Society of Naval Architects and Marine Engineers. He became a member of the A.S.M.E. in 1893.

ANDREW WARD FOOTE

Andrew Ward Foote, founder of Foote-Burt Co., died at his home in Cleveland, Ohio, June 3, 1925, after a short illness resulting from a stroke which he suffered on May 7. Mr. Foote, who had lived in Cleveland for thirty-eight years, was born in Guilford, Conn., in 1865. He was apprentice with Pratt and Whitney Co., at Hartford, Conn., 1882-1886, then was in the drawing-room of that company and of the L.&N.R.R. for about eighteen months. After serving in various capacities with the L.&N.R.R. he went to Decatur, Ala., as assistant to the master mechanic, and after several years became general foreman of machine, car, and smith shops. In 1892 he founded the Foote-Burt Co., known internationally as builders of machine tools. Due to failing health he retired as president of that company and disposed of his interests in it in 1919, at the time of his retirement from active business life.

Mr. Foote was intensely interested in the advancement of vocational training in the public schools, and his efforts are said to have been responsible for much of the vocational guidance work being done in the Cleveland schools. Through the work of Mr. Foote the special drills required in the building of the Panama Canal were made possible, and were produced by the company he then headed. He also had served on the board of the Electric Controller & Mfg. Co.

For years Mr. Foote had taken an active part in the advancement of engineering. He had been a member of the Society since 1900.

ROGER MORSE FREEMAN

Roger Morse Freeman died suddenly of acute appendicitis on January 21, 1925, at thirty-three years of age. In ten years he had accomplished more in engineering design and construction than most men find opportunity for in a lifetime. The cost of structures completed under his design and supervision in this short time aggregated more than $25,000,000. The end found him driving ahead on the design and construction of a 15,000-horsepower hydroelectric plant near Oakdale, on the Tippecanoe River in Indiana, with more than 500 men on a difficult rush job of winter construction.

Mr. Freeman was graduated from the Massachusetts Institute of Technology with a B.S. in electrical engineering in 1914. His studies at the Massachusetts Institute of Technology had been supplemented by a season at the great German engineering school in Charlottenburg where he specialized in internal-combustion engines. His engineering training began at the early age of fifteen when he was a rodman in a surveyor's party for the New York Board of Water Supply. Next a college vacation was spent as assistant engineer in a construction camp back of the Sierras on the Los Angeles Aqueduct, and still another as a machinist in the marine-repair shops of the Atlantic Works in East Boston. Graduation from college was followed by a year and a half spent as assistant engineer for John R. Freeman on the design of large hydroelectric developments in the United States and Canada, and as assistant engineer for the Turner Construction Co. on reinforced-concrete-building construction.

In January, 1916, Mr. Freeman became construction engineer for the Chase Rolling Mills and Waterbury Manufacturing Co. of Waterbury, Conn., concerns then in the rush of war work. He designed and supervised the construction of several multi-story reinforced-concrete factory buildings so satisfactorily that when his employers decided to build a large machine shop for fabricating their own heavy machinery for working brass and copper tubes and plates, almost the

only specifications given him were: "We need a machine shop about 400 feet by 100 feet. Make it the best of its size in New England."

As this building neared completion Mr. Freeman was summoned into conference by the Secretary of the Navy and requested to visit a certain large steel works under construction and report on what was causing delay. The result of his report was that he was made supervising engineer in the Navy Department and put in charge of the completion of the plant. In six months, working three shifts through one of the worst winters of recent years, he had the plant ready to forge shafts and guns. His success in getting all hands to pull hard together was so marked that he was sent here and there to speed up construction wherever it lagged. As supervising engineer in the Navy Department Mr. Freeman's work included the design and construction of large steel plants for the Erie Forge and Steel Co., Erie, Pa., and the Pollack Steel Co., at Cincinnati; the gun-treatment plant at the Allis Chalmers Co., Milwaukee; dormitories for the General Electric Co. at Erie; and the design and construction of the U. S. Navy armor-plate and gun-forging plant at South Charleston, W. Va.

Mr. Freeman had hoped to continue in metallurgical structural lines, but, no openings being found in the depressed conditions of this industry following the war, he turned back to hydroelectric construction in which he had had experience after leaving college. Plans were under way for him to join forces with his father, John R. Freeman, at the time of his death.

He had been a member of the Society since before his graduation from college, and was also a member of the American Institute of Electrical Engineers, the American Society of Civil Engineers, and the American Association of Engineers.

CHARLES E. FRITTS

Charles E. Fritts, member of the Society since 1915, died on May 5, 1925. Mr. Fritts was born at Pottersville, N. J., in 1872, but lived in Kansas City, Mo., the greater part of his life. He was graduated from Nebraska University in 1896, having taken a special course in electrical engineering. The following year, he entered the service of the Metropolitan Street Railway Co. of Kansas City, as assistant to the chief electrician. Later he was made chief electrician, and when the companies were consolidated he became electrical engineer. He served in that capacity until 1919, when he identified himself with the United Light and Power Co. of Kansas, as chief engineer of the steam and water-power generating stations of the central part of the state, his headquarters being at Abilene. Mr. Fritts was also a member of the American Institute of Electrical Engineers, the American Electric Railway Association, and the Masonic Order.

WILLIAM MONTGOMERY GARDNER

William Montgomery Gardner, member of the Society since 1914, lost his life in an accident on the Mississippi River on May 8, 1925, near Memphis, Tenn. Major Gardner was also a member of the A.S.C.E., and was one of twelve members of that society who were drowned when an excursion steamer overturned.

Mr. Gardner was born at Augusta, Ga., in 1868. He began work as a rodman on railroad construction, then worked as instrumentman in the city engineer's office of Memphis. In 1890, he was instrumentman on hydrographic survey for the Mississippi River Commission, in 1891 was inspector on levee location and construction, and in 1893 became U. S. Assistant Engineer on levee work on the river, in charge of the

Yazoo Levee District and the Upper Yazoo and White River Levee District. In 1905, he became U. S. Assistant Engineer in charge of river improvement work.

JUSTIN GRIESS, JR.

Justin Griess, Jr., vice-president of The McMyler-Interstate Co. of Cleveland, died at Bedford, Ohio, on February 26, 1925. He was born in Cincinnati on August 6, 1874, and was a graduate of Purdue University in mechanical engineering in 1896. Upon graduation he and Robert W. Kaltenbach became associated as Cleveland representatives of several prominent manufacturers, branching out as consulting engineers and establishing the firm of Kaltenbach & Griess of Cleveland. The firm later organized the Interstate Engineering Co. of Bedford, Ohio, which subsequently combined with the McMyler Mfg. Co., forming the present McMyler-Interstate Co.

Mr. Griess belonged to a large number of civic, social, fraternal, and engineering organizations in the Cleveland district, in some of which he took a very active interest, giving much of his time to them. Mr. Griess had been a member of the Society since 1898.

PAUL H. GRIMM

Paul H. Grimm, one of the "Thirty-Fivers" of the Society, passed away at Glen Cove, Long Island, on Dec. 17, 1925.

Mr. Grimm was born in Germany on May 3, 1853. He attended high school there and subsequently received private instruction in mechanical engineering.

In 1869 Mr. Grimm migrated to America and received his first engineering experience upon the old Morris & Essex Railroad at Dover, N. J., for which he worked for two years. Following this he put in two years in charge of steam plants in Brooklyn and New York. From 1873 to 1883 Mr. Grimm was chief engineer for the Duryea Starch Works at Glen Cove, and in that town he made his home for the rest of his life. From 1883 to 1893 he served the National Starch Co. in the same capacity, and then, after a short time as engineer with the Glen Cove Mfg. Co., Mr. Grimm joined the E. R. Ladew Leather Works, with which he was connected until 1903. Since 1903 he had been construction engineer for Arbuckle Bros.

Mr. Grimm became a member of the A.S.M.E. in 1890.

CHARLES HANSEN

Charles Hansen, who became a member of the Society in 1924, died January 11, 1925, at Clearwater, Fla., where he had been since October on account of his health. Mr. Hansen was born at Phillipsburg, N. J., on October 20, 1887. He attended grammar school at New Haven, Conn., and in 1912 received the Ph.B. from Sheffield Scientific School, Yale University.

After graduation, he became junior engineer with the New York, New Haven & Hartford R. R. Co., assisting in the installation of equipment and machines at Cos Cob. Later, he was resident engineer in charge of all new installations and construction work in the vicinity of New Haven, Conn. In 1917, he was rejected for the Army on account of his health, and therefore obtained a position with E. I. Du Pont de Nemours & Co., as supervisor of production at Wilmington, Del., and Nashville, Tenn. At the close of the war, he became assistant superintendent in charge of production for the National Carbon Co., Inc., of Cleveland, Ohio.

BENGT W. M. HANSON

Bengt W. M. Hanson, machine-tool inventor and gage engineer of wide renown, died at his home in Hartford, Conn., on September 6, 1925.

The career of Mr. Hanson as an engineer was a remarkable one. He was born in Sweden in 1866 and at the age of sixteen left school to enter upon the rigorous apprenticeship which used to be in vogue there in the mechanical trades. Upon the completion of his apprenticeship he received an appointment as a draftsman in the shops of the Swedish government. At this time his inventive ability began to be recognized.

In 1890 Mr. Hanson came to the United States, going first for a short time to the Sturtevant Blower Works in Boston and to the Thomson-Houston Electric Works in Lynn. His first permanent location was with the Waltham Watch Co. where he remained for five years and designed many precision automatic watch machines which are in general use today.

In 1895 Mr. Hanson was attracted to the Pratt & Whitney Co. by the tool work which they were doing and started in at their plant in Hartford as a bench worker. He remained with them for twenty years, during which time he designed many machine tools for accurate production work, devised new and finer systems of gages, and perfected manufacturing processes.

In spite of the fact that Mr. Hanson had been under the handicap of learning a new language and new customs, it was not long before he became the vice-president and general manager of the concern. One of the largest single contracts they ever received was worked out under his direction and covered a complete government small-arms plant which was set up near Hankow, China.

Upon leaving Pratt & Whitney in 1915 Mr. Hanson was put in charge of machine-gun production at the Colt Armory, as a civilian member of the War Department Board. He served here with distinction and throughout the critical period of the World War he was vice-president and general manager of the Colt concern. He had much to do with the development of the Browning gun during this time.

When the war ended Mr. Hanson left the Colt Armory and established an experimental and consulting engineering office in Hartford. In 1920 Clarence E. Whitney, of the Whitney Manufacturing Co., joined Mr. Hanson and the enterprise became the Hanson-Whitney Co. Besides establishing this new industry in Hartford, he brought the S. K. F. Ball Bearing Co. to that city. He was consulting engineer and director of this latter concern.

Mr. Hanson was not only a prolific inventor, but he had the unusual faculty of being able to carry his inventions through from conception to finished product and to prosecute their manufacture with the utmost vigor. He contributed signally to the present perfection of the interchangeable system of manufacture in America and but few people have realized the importance of the part which he played in behalf of the United States during the World War.

Mr. Hanson was a public-spirited citizen of Hartford and belonged to many active organizations there. He became a member of the Society in 1900.

OTIS W. HARROD

Otis W. Harrod, engineer with the Sioux City Gas & Electric Co., died on June 29, 1925, in California, where he had gone on a trip for his health. Mr. Harrod was born at Perryville, Ohio, on January 30, 1883, and received his early education in the schools of that city. He later attended Ohio State University, from which he was graduated

in 1907, with the degree of M.E. in electrical engineering. After his graduation he spent three years in the South as electrician with various lumber and telephone companies.

In February, 1910, Mr. Harrod started in with the Sioux City Gas & Electric Co. as meter foreman and it may be said that he grew up with this public utility. He was made assistant engineer in 1915 and as such carried out the construction of the present boiler and turbine plants, with the attending condensing and switchboard systems, etc. From 1918 to the time of his death Mr. Harrod was engineer of the company with full charge of production and distribution of both gas and electricity.

He became a member of the Society in 1921, and was also a member of the National Electric Light Association, the American Gas Association, and the Engineers' Club of Sioux City.

ROBERT D. HASSAN

Robert D. Hassan, designer and builder of many Portland cement plants, died at Celina, Ohio, on September 15, 1925. Mr. Hassan was born in Glasgow, Scotland, January 1, 1851. He served a seven years' machinist's apprenticeship there in the shops of Neilson Bros. In 1874, after completing this apprenticeship, he emigrated to America, finally settling in Cleveland where he was employed in the tool room and the drafting room of the Union Screw Co. From 1876 to 1893 Mr. Hassan was successively foreman of the Paper Box Machine Co. and the Brush Electric Co., and superintendent of the Short Electric Railway Co. He entered the Portland cement industry in 1893 as engineer and superintendent of the Sandusky Portland Cement Co., at Baybridge, Ohio. After four years with them he spent five years as engineer and superintendent with several cement concerns in Michigan.

Mr. Hassan in 1902 became a consulting engineer on cement manufacture and as such he designed and erected six large mills in Kansas, Michigan, and Canada. He contributed signally to the development of the rotary kiln and to the application of pulverized coal as fuel therein. Mr. Hassan was always interested in fine machine work and in this connection he had a part in the development of the Cleveland motor cycle. He retired from active work in 1922 and went to Celina, Ohio, where he bought a small machine shop. Here he found his recreation in light machine work and model making. Mr. Hassan was a member of the Masonic Order. He joined the Society in 1903.

WILLIAM CLARK HAWKINS

William Clark Hawkins, one of the best known power-plant engineers and executives in Canada, died at Hamilton, Ont., on May 11, 1925. Mr. Hawkins was born at Orange, N. J., on September 5, 1866. He was graduated from Stevens Institute in the class of 1888. After his graduation he gained a broad experience within a few years, working successively for the Campbell Printing Press Co., the Third Avenue Cable Railway in New York, and upon the erection of coffee machinery in Central America. His first major position was as manager of the Plymouth Electric Light Co., a General Electric subsidiary. Mr. Hawkins went to Hamilton, Ont., in 1901 to become managing director of the Hamilton Electric Light & Cataract Power Co. This later became the Dominion Power & Transmission Co. and with this company he remained until his death.

Mr. Hawkins was also president of the Electric Bond & Share Co. of Hamilton, and president of the Southern Canada Power Co. of Mon-

treal, whose power development on the St. Francis River he carried out. He was a director of several other power companies. Mr. Hawkins joined the Society in 1891.

ELWOOD HAYNES

Elwood Haynes, pioneer automobile inventor and manufacturer, died of pneumonia at his home in Kokomo, Ind., April 13, 1925. Mr. Haynes was one of the best known figures in the automobile industry, and was also known internationally for his work as a metallurgist.

Besides being the inventor and builder of the first successful automobile in America, he contributed a number of important inventions and improvements to the automobile. These included successful carbureters, mufflers, the use of aluminum in internal-combustion-engine construction and the adoption of nickel-steel and other alloys. His first successfully operated automobile is in the Smithsonian Institution at Washington, D. C.

In his metallurgical work, he made various alloys of cobalt, chromium, molybdenum and tungsten, the best known being the alloy Stellite, which is extensively used for lathe tools. His discovery of this alloy resulted from his search for a metallic combination that would resist the oxidizing influences of the atmosphere and still preserve a good cutting edge. In 1915, he became president of the Haynes Stellite Co., formed for the manufacture of this alloy.

Mr. Haynes was born on October 14, 1857, at Portland, Ind. He received the degree of bachelor of science from Worcester Polytechnic Institute in 1881, and, after teaching school for three years, spent a year studying chemistry and biology at The Johns Hopkins University. During the year 1885-86, he taught science at the East Indiana Normal School, and following the discovery of natural gas at Portland in 1886, he became manager of the Portland Natural Gas & Oil Co. From 1890 to 1901, Mr. Haynes was field superintendent for the Indiana Natural Gas & Oil Co., with headquarters at Kokomo. While there he became interested in gas engines and began studying the possibilities of applying the internal-combustion engine to light road vehicles.

On July 4, 1894, he successfully tried out his "horseless carriage," on which he had been experimenting for two years, and in 1898 he formed the Haynes Automobile Co. for the manufacture of automobiles, and continued as president of the company throughout the remainder of his life.

Mr. Haynes became a member of the Society in 1919, and belonged to a number of other engineering and scientific societies, including the American Institute of Mining and Metallurgical Engineers, the American Chemical Society, and the Iron and Steel Institute. He was an ardent advocate of the use of the metric system.

LAWRENCE R. HESSELINK

Lawrence R. Hesselink, die designer for the Steel Car Forge Co., Ellwood City, Pa., died at his home there on April 29, 1925. He was born at Arnhem, Holland, in August, 1886, and received his early education in that country. In 1903 he served a year's apprenticeship as a machinist in the shop of J. V. Harte in Deventer, Holland. After this he went to Switzerland and entered the Federal Polytechnic University at Zurich, where he completed his course in mechanical engineering in 1909. He remained for a year in Zurich as draftsman for Henry Baer & Co. on textile machinery, and then came to America, where he first went to work as a draftsman with the Lackawanna Steel Co. at Buffalo.

Between 1911 and 1920 Mr. Hesselink was successively employed in design work by the Kempsmith Mfg. Co., Milwaukee, Wis.; the Algoma Iron Works, Sault Ste. Marie, Ont., Canada; Gustave Lindenthal, New York, N. Y.; the American Brass Co., Waterbury, Conn.; the American Steel & Wire Co., Donora, Pa.; and the Standard Engineering Co., Ellwood City, Pa. Also, in 1913, he spent several months in Switzerland designing special machine tools for punching, planing, milling, and sewing asbestos cement. Mr. Hesselink became an associate member of the Society in 1919.

M. L. HOLMAN

M. L. Holman, retired mechanical engineer and former water commissioner of St. Louis, Mo., died on January 4, 1925, at his home in that city. Mr. Holman was born in Oxford County, Me., on June 15, 1872. He was graduated from Washington University in 1874 with the degree of A.B. The honorary degree of A.M. was conferred on him by his alma mater in 1905.

He began his professional career in the offices of the superintendent of architecture of the United States Treasury Department. In 1877 he became an assistant engineer in the St. Louis Water Works, and ten years later was appointed commissioner, in charge of the entire water system of the city. Mr. Holman immediately took up the work of extending and modernizing the system, the city expending for this purpose about $10,000,000.

During the twelve years that he held this office, he caused to be installed a new low-service pumping plant and six large settling basins, and extended an intake to the main channel of the Mississippi River. He installed new pumping stations, equipped with up-to-date machinery, which saved the city 80 per cent of fuel. He introduced the system of paying a bonus to builders for high efficiencies in pumping engines. It was under Mr. Holman's administration as commissioner that a division of the water distribution was made, so that a part of the city was supplied from one point and the remainder from another. He also inaugurated experiments in water sedimentation and purification by the aid of chemicals. In 1897 he visited Europe for the purpose of making a study of foreign filtration systems.

He served on the Board of Public Improvements of St. Louis, which has charge of the public engineering and structural work of the city, and as Chairman of the Committee on City Lighting had charge of the undergrounding of the electric wires, and the preparation of the specifications and letting of contracts for the lighting of streets, alleys, and public buildings.

Mr. Holman's success in his administration of the St. Louis Water Department established him as an authority in this field, and he was often called for consultation regarding similar problems in other cities. He served on the Board of Appraisers of Denver, which determined the value of the water system of that city and fixed schedules of water rates.

Upon resigning from the public service Mr. Holman was for five years general superintendent of the Missouri Edison Electric Co., and in 1904 became senior member of the firm of Holman & Laird, consulting hydraulic and mechanical engineers of St. Louis. This enterprise was abandoned upon Mr. Holman's retirement.

Mr. Holman was actively interested in the A.S.M.E., serving as vice-president of the Society in 1894-1896 and 1903-1905, and president in 1908-1909. He was also a member of the American Society of Civil Engineers and vice-president of that organization from 1905 to 1906. He was an honorary member of the American Water Works Association and belonged to the Engineers' Club of St. Louis.

WILLIAM D. HOXIE

William D. Hoxie, vice-chairman of the Babcock & Wilcox Co., died suddenly from heart trouble on January 12, 1925, on board the S. S. *Southern Cross* about three days north of Rio de Janeiro. He had started out on January 3 on an outing to South America, accompanied by his wife and niece.

Mr. Hoxie was born in Brooklyn on July 1, 1866, and received his early education in the public schools of that city. In 1889 he was graduated from Stevens Institute of Technology as a mechanical engineer. That same year he became connected with the Babcock & Wilcox Co., and all his active life thereafter was spent in the service of that company.

Early in his career Mr. Hoxie became connected with the marine department of Babcock & Wilcox and it was due largely to his energy and activity that installations of the Babcock & Wilcox boiler were made on board a number of merchant vessels.

In 1899 the first boilers designed by him were built for the U.S.S. *Alert*. This became the standard type of Babcock & Wilcox marine boiler, with the essential features of the drum in front, the increase of furnace volume toward the back of the boiler, and the baffling at right angles to the tubes. With slight modifications and improvements this is still the standard.

Mr. Hoxie was an early advocate of the use of superheat. He had a superheater fitted to the boiler of his yacht and ran a great many experiments, including one series under the direction of naval engineers which for a long time gave more reliable and useful information about superheat than could be obtained elsewhere.

One demand from the Navy met by Mr. Hoxie was for a very light boiler which could be secured only by tubes of 1 in. in diameter. After studying all the designs on the market he decided that the White-Forster boiler was the best for this purpose. It had the special merit of the ability to withdraw and replace a single tube without disturbing any other. The first boilers built by his company of this type had lower drums with the tube sheet flattened in accordance with the original design. Experience showed that there were objections to this type and Mr. Hoxie developed a cylindrical lower drum which removed the objection. This involved slight changes elsewhere in the boiler to retain the desirable feature of the removability of tubes. More than five hundred boilers of this type have been supplied for the U. S. Navy.

Mr. Hoxie's abilities, which were not only as an engineer but as a negotiator as well, marked him for high executive positions. In 1897 he became vice-president of the Babcock & Wilcox Co. In 1919 he became president of the company and at the time of his death had been made vice-chairman of the Board of Directors. Although his principal attention was always devoted to the marine department, he was constantly consulted on other engineering questions and matters of policy.

He was an ardent yachtsman and in recent years his yacht *Idalia* has been a center from which he dispensed princely hospitality to his many friends. He had fitted oil fuel to the boiler of this vessel and was enthusiastic about its success. He was a trustee of Stevens Institute of Technology, of Webb Institute of Naval Architecture, and of the Wilcox Memorial Library in Westerly, R. I.; a member of the Engineers, New York Yacht, and Lawyers Clubs of New York City, and of the Army and Navy Club of Washington; a member of the American Society of Naval Engineers, the Society of Naval Architects and Marine Engineers and the Chamber of Commerce of the State of New York, and a member of the A.S.M.E. since 1899.

HENRY W. JACOBS

Henry W. Jacobs, of Coloma, Mich., died on January 6, 1925, at Atchison, Kan. Mr. Jacobs was born at Atchison in 1874. He received his early education at the parochial school of Trinity Lutheran Church in Atchison, and when he was fourteen years old became a machinist apprentice under his uncle in the Seaton foundry. For two years, 1895-1897, he was in the United States marine service, then spent one year with the Sprague Electric Co. erecting elevators and other electric machinery. In 1900, he entered railroad work as an erecting machinist for the B. & M. R. R.; in 1903, became general shop demonstrator for the Union Pacific Railroad, engaged in standardizing methods of manufacture and tools. In September, 1904, he became engineer of shop methods for the Santa Fé System and in 1906 was appointed assistant superintendent of motive power of that railroad. In 1912 Mr. Jacobs left the Santa Fé to become president of the Oxweld Railroad Service Co., at Chicago. He maintained his interest in that company until his death, although he had resigned the presidency some years before on account of ill health.

Mr. Jacobs was an inventor, and had obtained sixty-eight patents in the United States. He introduced an improved oxygen-welding system in the Santa Fé shops and installed the system for other railroads. Mr. Jacobs became a member of the Society in 1905.

JOSIAH F. JOHNSON

Josiah F. Johnson, manager of the general engineering division of the South Philadelphia Works of the Westinghouse Electric & Manufacturing Co., died at the Chester Hospital on November 12, 1925. Mr. Johnson was born on July 15, 1879, at Zion Grove, Pa., and was graduated from both the high school and the normal school at Bloomsburg. After spending some time as a telegraph operator and as a school teacher, he entered Pennsylvania State College, from which he was graduated with honors in 1905.

Mr. Johnson joined the Westinghouse Machine Co. at East Pittsburgh in 1905 as a shop tester of steam turbines. After a year of this he was promoted to service work, and in 1910 to field engineer. He was successively assistant to district engineer in New York City and Rocky Mountain district engineer with headquarters at Denver. In 1912 he was recalled to the East Pittsburgh plant to become first assistant engineer of the steam-turbine department. Subsequently he became engineer of the turbine department and in 1923 he assumed his duties at Philadelphia. Mr. Johnson became a member of the Society in 1916.

HORACE KIMBALL JONES

Horace Kimball Jones, consulting engineer for the Corbin Screw Division of the American Hardware Corpn., died at his home in Hartford, Conn., on May 5, 1925, after a short illness. Mr. Jones was born at Stratton, Vt., in 1837. As a boy, he attended the village school in South Shaftsbury, Vt., later went to the Academy in North Bennington, and finally to Burr Seminary, Manchester. He was apprenticed to the Douglass Mfg. Co., at Arlington, Vt., from 1854 to 1857, then had drawing-room experience with John Hastings and shop experience with the Ball Machine Shop. In 1858, he became master mechanic for the Douglass Mfg. Co. and from 1863 to 1874 was master mechanic and superintendent for the Hart Mfg. Co. and the Hart, Bliner & Mead Mfg. Co. In 1874, he began work for the Corbin Screw Corpn. as mechanical superintendent.

Mr. Jones was the inventor of the screw machinery that was the foundation plant of the Corbin Screw. This machinery, invented in

1872 for the manufacture of wood screws, was set up by him in co-partnership with Samuel Hotchkiss in Colt's Northwest Armory at Hartford under the firm name of Jones & Hotchkiss. After their plant was bought, in 1875, by the Russell & Erwin Mfg. Co., a factory was built in New Britain and the business, continued under Mr. Jones' management as mechanical superintendent, was constantly increased and improved by his inventions. The patent records during the first thirty years of his management show over sixty inventions to his credit, many of which are used in various parts of the world today. In 1879, Mr. Jones invented and built machinery for making machine screws. He began the manufacture of carpenter squares by cutting from the solid sheet, and invented two processes of non-oxidizing annealing. Some of his patents, especially the multiple blow header, are considered the best of their kind.

About 1887, Mr. Jones started a movement for the standardization of screw threads, which finally resulted in the appointment of a committee by the A.S.M.E. Mr. Jones was a member of that committee, and in 1907 presented a report that was adopted by the Society. In addition to his work on screw machinery, Mr. Jones planned and directed the construction of the reservoirs of the Mattabessett River, in the early '70s, and the increase of the water power by construction and enlargement of the dam, which permanently increased and improved all the water power of the river. Mr. Jones had been a member of this Society since 1896.

WILLIAM R. JONES

William R. Jones, plant engineer of the University of Pennsylvania, died at the University Hospital in Philadelphia on October 26, 1925. Mr. Jones, who was born in Philadelphia in September, 1875, was a graduate of the University of Pennsylvania, from which he received the degree of B.S. in mechanical engineering in 1899 and of M.E. in 1911. From 1899 to 1903 Mr. Jones served as draftsman and surveyor with the Philadelphia National Export Exposition, the Pennsylvania Iron Works, and the Philadelphia & Reading Railroad. Mr. Jones had been connected with the staff of the University of Pennsylvania since 1903. He was successively staff assistant in dynamics, director of light and power, engineer of construction, superintendent of buildings and grounds and plant engineer. He was a member of the American Society of Heating and Ventilating Engineers and the Engineers' Club of Philadelphia. Mr. Jones became a member of the A.S.M.E. in 1909.

CLARENCE E. KILBY

Clarence E. Kilby, chief engineer of The Engineer Co., New York, died on April 30, 1925. Mr. Kilby was born on September 29, 1874, in Highland, N. Y., where he attended grammar and high schools. Later he spent one year at Stevens preparatory school. In 1906, he was placed in charge of the engineering work of The Engineer Co. He was specially interested in the installations of balanced draft, furnace regulation, stoker control, and Turner baffle walls. Mr. Kilby became a member of the Society in 1921.

PROF. GEORG KLINGENBERG

Prof. Georg Klingenberg, noted electrical engineer and a Privy Councillor in Germany, died at Berlin on Dec. 7, 1925. Professor Klingenberg was born at Hamburg on Nov. 28, 1870, and received his preliminary education at the Realgymnasium at Osnabruck. He then

attended the University of Rostock, from which he received his degree of Doctor of Philosophy in 1895.

From 1895 until 1910 he served as professor at the Technisch Hochschule at Berlin and as a consulting engineer. In 1902 he was appointed a member of the board of the German General Electrical Co. Between 1907 and 1914 he was consulting engineer upon some of the great projects of this concern, including the Rosherville Dam of the Victoria Falls Power Co. in South Africa, and the Golpha Central Station, which at the time of its construction in 1915 was the largest steam generating plant in Germany. Professor Klingenberg made numerous researches resulting in the improved efficiency of steam plants, and also contributed much to the development of internal combustion engines.

He received high honors from various institutions and societies in Germany in recognition of his work and of his prolific writings upon engineering matters. The Technisch Hochschule of Berlin made him an honorary Doctor of Engineering in 1918, and in 1921 he was elected president of the Verein Deutscher Ingenieure.

Professor Klingenberg became a member of the A.S.M.E. in 1924.

AUGUST H. KRUESI

August H. Kruesi, until a year ago chief construction engineer of the General Electric Co. at Schenectady, N. Y., died May 7, 1925, at El Paso, Texas. Mr. Kruesi was born at Newark, N. J., in 1876 but went to Schenectady when he was very young, when his father went to that city with Thomas A. Edison to establish what has become the General Electric works. Mr. Kruesi received the bachelor's degree in electrical engineering from Union College in 1898, and M.E.E. in 1903. In 1898-1899, Mr. Kruesi was electrical engineer with the British Thomson-Houston Co. in London, and the following year was with H. F. Parshall, designing electric generators. In 1900, he began work as electrical and designing engineer with the General Electric Co., and in 1907 was made chief construction engineer. Two and one-half years ago he went to Texas for his health, and a year ago resigned his position. Mr. Kruesi joined the Society in 1901.

FRANK L. LEACH

Frank L. Leach, assistant manager of the Agricultural Implements Co., Ltd., died at Jamshedpur, India, on April 7, 1925, after four days' illness. Mr. Leach, who was born at Sandusky, Ohio, in 1891, obtained his engineering degree by studying at the Y. M. C. A. and taking International Correspondence School courses. He began work in field engineering with the East Ohio Gas Co. at Cleveland when he was seventeen years old and worked up through various positions with other companies until eleven years later he was assistant chief draftsman with Perin & Marshall, New York City.

After about two and one-half years in the Perin & Marshall offices, Mr. Leach left for India, January, 1922, to take charge of the construction of the plant, already started, for the Agricultural Implements Co., a subsidiary of the Tata Iron & Steel Co. Mr. Leach worked with his usual concentration on the installation of the plant and, after December, 1922, on its operation, and credit for a great part of its success is given to him. Mr. Leach became an associate member of the Society in 1919.

JAMES WILLIS LOUDER

James Willis Louder, long connected with the Portland cement industry at Nazareth, Pa., died at Atlantic City on March 7, 1925. Mr. Louder was born at Easton, Pa., on April 27, 1871, where he re-

ceived his preliminary education. After a short course in chemistry at Lafayette College, he went to work in 1895 in the laboratory of the Andover Furnace at Phillipsburg, Pa. A few months later he entered the cement business as chemist of the Atlas Cement Co., with which he remained for many years, eventually becoming assistant superintendent. Mr. Louder joined the Nazareth Cement Co. in 1907 as superintendent and at the time of his death was general manager. He became a member of the Society in 1914. He was also an active member of the Masonic Order.

CHARLES W. LUMMIS

Charles W. Lummis, manager of construction with the Gas Producer Department of the Morgan Construction Co. of Worcester, Mass., died on Dec. 28, 1925, at Worcester. Mr. Lummis was born at Cedarville, N. J., on Aug. 22, 1880. He was educated in the public schools of Philadelphia, followed by engineering courses by correspondence.

In 1898 he entered the drafting room of the Camden Iron Works. After one year there he joined E. A. W. Jefferies, the present manager of sales in the Gas Producer Department of the Morgan Construction Co., as assistant in various projects in Michigan and Colorado. After three years he became connected with the Wellman-Seaver-Morgan Co. of Cleveland. He carried on drafting and inspection work with them from 1901 to 1903. In 1903 he returned to the Camden Iron Works, working upon the problems of the design and operation of gas producer plants for power and heating purposes, and becoming mechanical engineer of this concern. He was also located in 1910 at the Scoville Mfg. Co., where he worked on the problems of casting brass.

In 1913 Mr. Lummis rejoined his former associate Mr. Jefferies, this time at the Morgan Construction Co., where he became Mr. Jefferies' assistant in the Gas Producer Department. He held the position of manager of construction up to the time of his death, and was considered one of the leading engineers in this field. He joined the Society in 1907.

WILLIAM H. MADDOCKS

William H. Maddocks, who, until his retirement in 1921, was assistant superintendent of machinery and equipment of the Missouri, Kansas & Topeka Railroad, died suddenly on February 27, 1925. Mr. Maddocks was born on December 1, 1861, in England, where he received his early education. Later, in this country, he attended Cornell University for three years, taking the mechanical engineering course. He served a six-year apprenticeship as machinist and worked as a journeyman for eight years. He was in succession draftsman, foreman and general foreman of the Missouri & Pacific Railroad, then became chief draftsman of the machinery department of the Missouri, Kansas & Topeka Railroad. Later he was appointed assistant superintendent of machinery and equipment for the same road. He resigned in 1921 because of ill health. Mr. Maddocks joined the Society in 1891.

FRANK L. MASSEY

Frank L. Massey, president and treasurer of the Massey Machine Co. of Watertown, N. Y., died in that city on April 21, 1925. Mr. Massey was born at Cleveland, Ohio, on March 10, 1872. His education was obtained in the public schools of Cleveland and of Watertown, where his family located during his boyhood. He supplemented this by evening study in mechanical engineering. Mr. Massey's first practical experience was gained in the shop and drafting room of

the New York Air Brake Co., where he served his apprenticeship from 1888 to 1893. From 1893 to 1898 he served as foreman in several of their departments, and from 1898 to 1901 was assistant superintendent in charge of the shops. In 1901 the company sent him to Moscow, Russia, as engineer in charge of construction and equipping of a new plant there. Mr. Massey returned to the United States in 1903 and organized the Massey Machine Co. From that time until his death, Mr. Massey specialized in the design and installation of engine governors, being recognized as an authority in this subject by the largest engine builders. He became a member of the Society in 1916.

ASA MARTINES MATTICE

Asa Martines Mattice, a member of the Society since 1889 and a manager from 1903 to 1906, died at the Engineers' Club, New York, April 19, 1925.

Mr. Mattice was born in Buffalo, N. Y., August 1, 1853. His reputation began with his record at the Naval Academy, where he led his class. He entered the Academy with the second class of Cadet Engineers, for the two-year course. He was graduated in 1874 and was at sea on the *Brooklyn*, *Vandalia*, and *Trenton* until the summer of 1879.

In September, 1879, he was ordered to duty at the Naval Academy, where for the next three years he was an instructor in the Department of Steam Engineering.

In 1882 he was ordered to sea on the *Miantonomoh* and the next year was transferred to the *Juniata*, which was the first foreign ship to bring succor after the eruption of Krakatoa, in the Sunda Straits, in which about half of an island was blown to atoms. She remained in the vicinity for months, taking soundings, directing shipping, making new charts, etc., in all of which Mattice had a leading part.

At the end of this cruise, he was ordered to duty in the Bureau of Steam Engineering, where he remained, except for a few months of detached duty, during the rest of his active service in the Navy. This was at the beginning of what is generally called in the Service the "New Navy." Chief Engineer, afterwards Admiral, Melville was appointed senior inspector at the Cramp Works, where several naval vessels were being built. Plans for some of these had been obtained abroad, but Melville, satisfied that he could produce equally good or better designs, obtained authorization from Secretary Whitney to prepare some designs. As one of his assistants by whom all the work of calculation and drafting was done, Mattice, with his tireless energy, his swift work, and his vast fund of information, played an important part in preparing the design which came into actuality as the machinery of the *San Francisco* and was the earliest of Melville's many successes in the design of machinery for our naval vessels.

Mattice was chief designer and in charge of the drafting room of the Bureau of Steam Engineering from August, 1887, to the end of December, 1888, when he went on leave for a year. He produced the design for the machinery of the original *Maine* (blown up in the harbor of Havana in 1898) and in connection therewith prepared the machinery specifications. These were the most elaborate and complete specifications made for an installation of marine machinery up to that time, and served for many years, with necessary modifications, as the model for specifications.

When he left the Bureau of Steam Engineering in 1890, Mattice joined the staff of E. D. Leavitt of Boston, as principal assistant. He had full charge of all designing, and turned out plans for hoisting and pumping engines and mining machinery that remain unsurpassed for both quality and size.

After about ten years of splendid work with Leavitt, Mattice left him and was occupied in consulting work alone for about a year, when he joined the Westinghouse Electric Co. as chief engineer. At the end of three years, Mr. Westinghouse transferred him to the Westinghouse Machine Co. as chief engineer, but retained his connection with the Electric Company as consulting engineer.

Early in 1904, Mattice's old friend and classmate, B. H. Warren, became president of the Allis-Chalmers Co., and urged Mattice to go with him as chief engineer and in charge of manufacturing, which he did in April of 1904. Together, he and Charles C. Tyler, the general superintendent, planned the large extensions of the Allis-Chalmers works at Milwaukee.

After two years of successful service with the Allis-Chalmers Co., Mattice resigned to become a partner in the newly organized firm of consulting engineers, Kafer, Warren and Mattice, with offices in New York City. Mr. Kafer and Mr. Warren both died within a year after the organization of the firm, and their death so affected Mattice that he felt unable to continue the business alone. In 1907 he became works manager of the Walworth Manufacturing Co. of Boston, where he had full charge of the design and manufacture.

In 1911 he retired from active business, but in 1915, at the earnest solicitation of Mr. Tyler, who had become vice-president of the Remington Arms Co., Mr. Mattice joined that company as advisory engineer. After the Armistice, he prepared most of the data required by the company for the settlement of its war contracts with the United States and other governments, and also in connection with its several government suits.

During 1917 and 1918 Mattice was secretary of the Small Arms and Ammunition Manufacturers Production Committee, whose members comprised representatives from each of the manufacturers of these products, who had contracts with the United States Government.

In addition to being one of the early members of this Society, Mr. Mattice was also an original member of the Society of Naval Engineers and a member of the Engineers' Club of New York.

JOHN A. McGREGOR

John A. McGregor, of Athol, Mass., died on March 26, 1925, after a long illness. Mr. McGregor was born on August 11, 1856, in Brookville, Nova Scotia, Canada. He started on his career in 1872 with the Brown & Sharpe Mfg. Co., Providence, R. I., where he learned the machinist's trade. He remained with that concern for twenty years, then resigned to become vice-president of E. Harrington Sons & Co., Philadelphia, Pa. Two years later, he became general superintendent of the Morse Twist Drill & Machine Co., New Bedford, Mass. In 1905, Mr. McGregor, together with a number of associates, purchased the former Gay & Ward business in Athol, Mass., now known as the Union Twist Drill Co., of which he was president at the time of his death. He was also a director of the L. S. Starrett Co. of Athol. Mr. McGregor became a member of the Society in 1902. He was very much interested in the civic affairs of Athol and for a number of years was president of the Worcester Northwest Agricultural Society. He was well known as an enthusiastic horseman and his horses raced on many tracks under his colors.

JAMES A. MEASE

James A. Mease, associate professor of mechanical engineering at Lehigh University, died on Dec. 21, 1925, at the home of his parents in Sidersville. Professor Mease, who was born at Bethlehem, Pa., on

Sept. 7, 1882, was a graduate of the Bethlehem High School of Lehigh University, where he received the degree of M. E. in 1905.

After graduation he spent seven months with the Dupont Powder Co., Wilmington, Del., on design and tests of their power plants. This was followed by ten months in the design of equipment with the Latrobe Steel Co. at Chicago. In 1906 and 1907 he designed blast and open-hearth furnaces for the Tennessee Coal, Iron & Railroad Co., in Birmingham, Ala. He also spent about two years with the National Tube Co., Latrobe, Ohio, in charge of rolling-mill design.

From 1909 to 1917 Professor Mease was at Pennsylvania State College, in the Department of Mechanical Engineering. Here he was successively instructor and assistant professor, and was in charge of the courses in machine design. He left there in 1917 to take charge of the Engineering Department of the John W. Cowper Co. of Buffalo, N. Y. This work was concerned with the design and construction of power plants and also mechanical equipment for large buildings. He was also for a short time engaged in similar work with the F. L. Grant Co. of Clarksburg, W. Va.

Professor Mease became a member of the faculty at Lehigh University in 1921, as assistant professor of mechanical engineering. In 1922 he was promoted to associate professor and at the time of his death was in charge of the courses in machine design. Among his writings were many articles in technical journals, and a book on Elements of Machine Design. He had belonged to the A.S.M.E. since 1913.

GEORGE MESTA

George Mesta, president of the Mesta Machine Co., Pittsburgh, Pa., died on April 21, 1925. Mr. Mesta was born in 1862, in Bethel township, Allegheny County, Pa., where he received his early education. He was graduated in 1885 as a civil engineer from the University of Pittsburgh. For two years he was associated with Totten & Co., as engineer, designing engines and rolling-mill machinery. In 1887 he organized the Leechburg Foundry & Machine Co. and became vice-president and general manager. In 1898, he founded the Mesta Machine Co., of which he was president to the time of his death.

Mr. Mesta became a member of the Society in 1894. He belonged also to the American Iron and Steel Institute, the American Institute of Mining and Metallurgical Engineers, the National Association of Manufacturers and Engineers, the Society of Western Pennsylvania, and to the Machinery, Engineers' and Bankers' Clubs of New York as well as to various civic and social organizations.

GEORGE H. MILLER

George H. Miller, supervisor of engineering for Cheney Brothers, silk manufacturers, of South Manchester, Conn., died in a hospital in Morristown, N. J., on October 5, 1925. He was born on October 22, 1870, at Washington, Me., and was a graduate of Worcester Polytechnic Institute, with the degree of B.S., in 1892. He began work as a draftsman with the Merrick Thread Co., Holyoke, Mass., and until 1898 was engaged upon their special machine designing and building. In 1898 he became chief draftsman for Tower & Wallace on paper-mill construction. He joined the American Thread Co. at Holyoke, Mass., in February, 1899, as local engineer of this plant. For the past fifteen years Mr. Miller was located at Manchester, Conn., having charge of mill construction and maintenance, and special machine design. Mr. Miller's health broke down during the World War on account of his overwork on various drives and as local Food Administrator, and he never fully recovered. He had been a member of the Society since 1906.

CHARLES MILLS

Charles Mills, consulting engineer with the Saco-Lowell Shops, died at the Brooks Hospital in Brookline, Mass., on July 4, 1925. Mr. Mills was born at Rochdale, Lancashire, England, in 1852. He went to work very early in life, his first mechanical experience being gained in his father's shop in the town of Heywood, England. He later was employed by the engineering firm of Howard & Bullough, and was sent by them to India, where he remained for three years. Mr. Mills came to the United States over thirty years ago and for most of this period he was connected with the Saco-Lowell Shops as inventor, works manager, and engineer. His inventions in the field of textile machinery were numerous and are in operation the world over. He concentrated his energies especially upon the improvement of the revolving flat card. Mr. Mills was deeply interested in the work of the Masonic Order. He joined the Society in 1907.

JOHN Z. MURPHY

John Z. Murphy, electrical engineer of the Chicago Surface Lines, died at Phoenix, Ariz., on January 16, 1925. Mr. Murphy had a remarkable career. He was born in a log cabin in an obscure village in northern Illinois in 1853, and was forced to leave school at an early age to support his widowed mother. His first job was that of tow boy on the old Illinois Canal. He worked up as railroad laborer, derrick fireman, and marine engine oiler to be a licensed steam engineer aboard a lake-steamer at the age of twenty-one. He then turned from steamships to railroads again, this time in the construction and civil engineering field with the Chicago and Northwestern Railroad. For five years he was engaged in this work and achieved fame in the bridging of the Platte River. After this he turned again to steamships and spent six years as chief machinist for the Morgan Line. As representative of the Pennsylvania Iron Works he entered the traction field when he installed the cable machinery of the West Chicago Street Railroad.

After the installation he remained with the railway as chief engineer, later being made chief engineer of the Chicago Railways Co. By the time electricity superseded the cable Mr. Murphy had trained himself in the new power and superintended the installation of it. When the Chicago lines were unified in 1914 he was made electrical engineer of the entire system, and held this position up to the time of his death. Mr. Murphy became a member of the Society in 1899.

ERNEST WILLIAM NEWMAN

Ernest William Newman, manager for Kieley & Mueller, Inc., died on April 11, 1925. Mr. Newman was born on March 6, 1865, at Newark, N. J., and was educated in the public schools of Newark and New York. He began his work in 1881 as a machinist's helper at the Pennsylvania R. R. shops at Meadows, N. J. After experience with several firms as helper, steam-fitter, and foreman, he became vice-president and general superintendent of the Frank Dobson Co., in 1898. He retained that position until 1912, when he started his own business. Six years ago, he began work with Kieley & Mueller, manufacturers of steam specialties. Mr. Newman became a member of the Society in 1921. He was an Odd Fellow.

CONRAD NORDFORS

Conrad Nordfors, chief engineer of Walter Scott & Co. of Plainfield, N. J., died in that city on August 27, 1925. He was born at Stockholm, Sweden, on November 12, 1870. His education was obtained there, and

he was graduated from the Royal Polytechnical School in 1889. Soon after his graduation he came to America, and was engaged in contracting work on the New York street railways from 1890 to 1893. He then spent four years as draftsman with several small manufacturing concerns in New York City.

In 1897 Mr. Nordfors went to work for R. Hoe & Co. designing special tools used in printing presses. He remained with them for six years and during that time was the designer of considerable new printing machinery. From 1903 to 1909 he was engaged in the design of photoengraving and stereotyping machinery for F. Wesel Mfg. Co. He was with the Wood Newspaper Machinery Corpn. of New York from 1909 to 1914 in charge of design, construction and erection of high-speed newspaper machinery. From 1914 to 1920 Mr. Nordfors was chief engineer of this concern on outside work, such as laying out press rooms, installing machinery, and handling contract work. He left it to take up the work in which he was engaged at the time of his death. He became a member of the Society in 1919.

LOUIS C. NORDMEYER

Louis C. Nordmeyer was born in Covington, Ky., in 1869. He supplemented his common-school education by home study and evening courses in the public schools of St. Louis, Mo., and the art school connected with Washington University. He worked as a machinist in St. Louis for two years, then took up the work of operating engineer in electric-light and power plants. In 1893, he entered refrigerating engineering work as an operator with the East St. Louis Ice and Cold Storage Co., and in 1897 became chief engineer of the Griesedick Artificial Ice Co. in St. Louis.

In 1900, Mr. Nordmeyer took up the work of designing refrigerating plants and installation of machinery. He became sales engineer in the St. Louis territory for the Frick Co. of Waynesboro, Pa. At the end of two years, he was transferred to the general office and factory at Waynesboro, Pa., and was given the position of mechanical engineer at the plant. After several years of work at the shops, he returned to St. Louis and again took charge of that territory for the Frick Co.

The Tait-Nordmeyer Engineering Co., consulting engineers, was incorporated in 1907, the other member of the firm being Mr. R. H. Tait. Mr. Nordmeyer was secretary and treasurer of the company. The work of the firm consisted chiefly in designing and installing complete refrigerating plants, buildings, and equipment. In 1915, and again in 1917, Mr. Nordmeyer went to China in the interest of an American concern, to superintend the construction of buildings and installation of machinery of the first all-American industry in China for freezing and drying eggs. In 1922, Mr. Nordmeyer disposed of his interest in the Tait-Nordmeyer Engineering Co. and with his son, Carl L. Nordmeyer, organized the Nordmeyer Engineering Co.

Mr. Nordmeyer became ill soon after he returned in 1920 from his last trip to China, and never regained normal health. After January, 1923, he spent many months in hospital, and received about fifteen blood transfusions in an effort to recover from pernicious anemia. He died on April 2, 1925.

In 1899, Mr. Nordmeyer became a junior member of the A.S.M.E., and in 1913 was advanced to full membership. During the years 1920-1923, he was a manager of the Society. He was very active in the Local Section work of the Society and during the past year served as chairman of the St. Louis Section. As chairman of the local committee in charge of the 1920 Spring Meeting, he was responsible for a meeting that stands out for a fine exhibition of hospitality and careful manage-

ment. In 1920, also, he was a delegate representing the A.S.M.E. at the Washington Conference that evolved The Federated American Engineering Societies.

Mr. Nordmeyer was a charter member of the American Society of Refrigerating Engineers, was a director in 1913-1914, and was always active in the Society's affairs. He also belonged to the Engineers' Club of St. Louis.

HENRY M. NORRIS

Henry M. Norris, who was chairman of the Cincinnati Section in 1920, and a manager of the A.S.M.E., 1920-1923, died suddenly of heart failure at Cincinnati on Dec. 24, 1925. Mr. Norris was prominent in machine tool circles as secretary of the Cincinnati Bickford Tool Co., and was active in his work up to the moment of his death.

Mr. Norris was born at Trenton, N. J., on Jan. 21, 1868. He received his early education at the State Normal and Model Schools, Trenton, Trenton Academy, and the Lawrenceville (N. J.) School. While still attending school he served an apprenticeship in the shops of Bement, Miles & Co., at Philadelphia, and subsequently he spent a year in the drafting room of the Phoenix Iron Co., at Trenton. He then studied mechanical engineering at Sibley College, Cornell University. After attending Cornell he was draftsman at the Ferracute Machine Co., at Bridgeton, N. J., for the Brown & Sharpe Mfg. Co., at Providence, and for the Pond Machine Tool Co. at Plainfield, N. J. In 1894 and 1895 he was superintendent and engineer for the Appleton Mfg. Co. and Riehle Bros. Testing Machine Co. in Philadelphia. In 1896 he became general manager of the Campbell & Zell Co., of Baltimore.

Mr. Norris joined the Bickford Drill & Tool Co., of Cincinnati, as a stockholder and works manager in 1897, continuing in this capacity until the formation of the Cincinnati Bickford Tool Co. in 1909, at which time he became secretary and a director of the new organization.

Mr. Norris was a prolific inventor in the field of machine tools, and was generally recognized as the pioneer designer of the modern type of radial drill. Among his improvements upon this machine may be mentioned the doubling of the number of speeds and feeds and means for shifting them instantly, accurate means of tripping the feed automatically at any predetermined depth, rapid arm adjustments, etc. He was one of the first to work out formulas for the power required to drive drills under various conditions. During the war Mr. Norris was a member of the Council of National Defense and served as a mechanical engineer for the Ordnance Department. He was a charter member of the Army Ordnance Association. He was an authoritative writer upon machine-tool subjects, shop management, and history.

In 1892 Mr. Norris joined the A.S.M.E. as a junior, and became a full member in 1898. He was also a member of the Engineer's Club of Cincinnati, of which he was a director from 1920 to 1923, the Cornell Club, the Loyal Legion, and many other patriotic and historical societies.

CHARLES H. PARKER

Charles H. Parker, superintendent of the generating department of the Edison Electric Illuminating Co. of Boston, died on January 28, 1925. Mr. Parker was born on February 6, 1873, in Hong Kong, China. He was graduated from the Massachusetts Institute of Technology in 1895 and entered the employ of the Edison Electric Illuminating Co. of Boston where in a few years he became superintendent of the generating department. During the Spanish War Mr. Parker served in the Navy, being retired at its close with the rank of captain. During the

World War he was Chief of Bureau of the Massachusetts Public Safety Council and a member of the Naval Sub-Base Committee.

Mr. Parker became a member of the Society in 1903. He belonged to the United States Naval Institute, the Edison Pioneers, the National Electric Light Association, the American Institute of Electrical Engineers, the Illuminating Engineering Society, the Engineers' Club of Boston, and the Masonic Order.

EDWIN J. PEIRCE, JR.

Edwin J. Peirce, Jr., founder and for years vice-president and general manager of the Taft-Peirce Manufacturing Co., Woonsocket, R. I., died on January 19, 1925. Mr. Peirce was born on January 25, 1852, in Providence, R. I., where he was educated. After serving an apprenticeship with the Brown & Sharpe Manufacturing Co., he was employed by that company as a draftsman. Later he worked for the Rhode Island Locomotive Works and the Harris-Corliss Steam Engine Works in the same capacity. From 1876 to 1888 he was connected with the Wardwell Sewing Machine Co., Woonsocket, and when the concern was reorganized as the Taft-Peirce Manufacturing Co., he became superintendent and later vice-president and general manager. He resigned from that position in 1907 to open an office in New York City as consulting mechanical engineer on machinery problems. From 1912 to 1916 he served as production manager for R. Hoe & Co. During the World War Mr. Peirce gave his services as production expert. Later he opened an office in New York City where for a number of years he employed a staff of consulting engineers. Mr. Peirce became a member of the Society in 1914. He belonged to the Masonic Order and to the Elks.

ROWAN P. PERKINS

Rowan P. Perkins, of the B. F. Sturtevant Co., died on Dec. 11, 1925. He was born at Rosemont, Pa., Aug. 24, 1884, and was graduated from the University of Pennsylvania in 1906 with the degree of B. S. in mechanical engineering. This was followed in 1909 by a degree of M. E.

From July, 1906, until November, 1912, Mr. Perkins was with the United Gas Improvement Co., of Philadelphia. Starting in as street clerk, he was promoted to foreman of fitters, to the Fuel Appliance Laboratory, and finally into the Construction Department.

He left the concern in 1912 to become a partner in the firm of Perkins, Fulweiler & Co. gas appliances, and in 1913 sold his interests to enter into the field of consulting engineering. This work he carried on successfully until 1922 when he entered the mechanical engineering department of the New York Central. He resigned this position Apr. 1, 1925, to become associated with the B. F. Sturtevant Co., Hyde Park, Boston, Mass. Mr. Perkins became an associate member of the A.S.M.E. in 1914.

WILLIAM A. PITTSFORD

William A. Pittsford, fuel combustion engineer for the Kewanee Boiler Co., Kewanee, Ill., died on April 17, 1925, after a short illness. Born in 1874, at Granville, Ohio, Mr. Pittsford was graduated from high school and obtained his engineering experience through practical work. He served the City of Chicago as mechanical engineer for eight years, then, in 1914, went to the Kewanee Boiler Co. as combustion engineer and designing engineer. Mr. Pittsford was a 33d degree Mason. He became an associate member of the Society in 1917.

CHARLES E. POPE

Charles E. Pope, consulting engineer, died on May 16, 1925, at his home in Cleveland, Ohio. Mr. Pope was born in 1867, at Cleveland, and was educated at the Case School of Applied Science and Cornell University, graduating from the latter with the class of 1892. Mr. Pope's business connections were with the Eberhard Mfg. Co., the National Malleable Castings Co. and the Thompson Electric Co. For some years before his death, he had his own office as a consulting engineer.

Mr. Pope became a member of the Society in 1897. He was a member of the Cleveland Engineering Society, and a 32d degree Mason. He was a Captain of Engineers, retired, a member of the Ohio National Guard, and a veteran of the Cleveland Grays.

HARRY W. L. PORTH

Harry W. L. Porth, master car builder for the Swift Refrigerator Transportation Co., at Chicago, died in that city on Mar. 2, 1925. He was born at Utica, Kan., on Jan. 18, 1889. While attending school he began his engineering career by work during vacations upon smelter construction and operation with the Balakalala Construction Copper Co. and upon the Yuba River irrigation project. During that time he was attending the Missouri University of Mines, from which he was graduated in 1911 with the degree of B. S. in mining engineering.

Before joining the organization of Swift & Company in 1912, he spent about a year as millman, assistant assayer, and refiner with the Nevada Hills Mining Co. Mr. Porth went to work for Swift & Company at Kansas City as draftsman and mechanical engineer. In 1914 he was promoted to assistant master mechanic and in 1916 became head of the planning department. He was transferred to the car department at Chicago as subforeman in 1917, and shortly afterward was made assistant master car builder. Upon the death of J. E. Libby, master car builder, in 1919, Mr. Porth succeeded to this position.

Mr. Porth was a member of many fraternal and technical organizations, including the Masonic Order, the Kappa Sigma and Tau Beta Pi fraternities, the American Welding Society, and the Western Railway Club. He became an associate-member of the A.S.M.E. in 1920.

CHARLES W. PUSEY

Charles W. Pusey, veteran marine mechanical engineer and one of the very early members of the A.S.M.E., died at Wilmington, Del., on Dec. 11, 1925.

He was a son of Joshua L. Pusey, who founded The Pusey & Jones Co. at Wilmington in 1848, and was born in that city on Aug. 20, 1843. He attended both public and private schools there, completing his formal education at the T. Clarkson Taylor Academy. While going to school he spent all his spare time in the shops.

In 1860, at the age of seventeen, Mr. Pusey began his engineering career in earnest as an apprentice in the Pusey & Jones plant. This was but a short time before the outbreak of the Civil War and the business was very dull. When the war began this condition was suddenly reversed as large United States contracts were received, covering machinery for naval vessels laid down at the Philadelphia Navy Yard. These government contracts continued to come in throughout the war and the young man rapidly gained broad experience in shop, drafting room, and in the field.

Shortly after the Civil War the company sent Mr. Pusey to San Francisco to install one of their engines in an ocean tug. Upon the

successful completion of this, one of the Pacific coast steamship companies obtained his services as their supervising engineer. This position he held for some time, resigning to become marine inspector for an eastern railroad company. He was later an inspector for the Quartermaster Department.

After an absence of eight years Mr. Pusey returned to The Pusey & Jones Co. and remained with them for the rest of his life. From 1891 to 1902 he served as president of the company. During the World War he was very active in the field of marine engineering and also carried on some important work for the Ordnance Department.

Mr. Pusey was a member of the Society of Naval Architects and Marine Engineers. He was a Life Member of the A.S.M.E. and one of the early members, having joined in 1880. From 1892 to 1895 he served as a Manager of the Society.

WILLIAM EVERETT QUIMBY

William Everett Quimby, president of Wm. E. Quimby, Inc., Newark, N. J., died on February 2, 1925. Mr. Quimby was born on March 1, 1866, in Orange, N. J. He was educated in the Dearborn-Morgan School and in the public schools of Orange. He was graduated from Stevens Institute of Technology in 1887 with the degree of M.E. Upon his graduation he studied law for a time in his father's office but afterward went into the manufacturing business. About thirty years ago Mr. Quimby invented a pump which is known as the Quimby pump, and organized a company—William E. Quimby, Inc.—for its manufacture. He was president of this concern at the time of his death. Mr. Quimby became a member of the Society in 1889. He was a member of the Engineers' Club of New York. He was particularly interested in civic affairs and in the successful working of the Welfare Federation of the Oranges.

EDWARD S. REID

Edward S. Reid, member of the Society since 1896, died at his home at Detroit, Mich., on June 13, 1925, after a short illness. He was born in Manhattan, Kan., April 18, 1871. After the death of his father, Rev. Jos. Reid, in 1877, the family located at Romeo, Mich. Mr. Reid was graduated from the elementary schools and from the high school at Vassar, Mich., in 1889. He entered the University of Michigan the same year and was graduated with the mechanical engineering class of 1893.

Immediately upon graduation, Mr. Reid was associated with Dean Mortimer Cooley in special engineering research work, after which he accepted a position with the Whiting Foundry & Equipment Co., of Harvey, Ill., in September, 1894. He continued with this organization until 1898, when he collaborated with George True and Henry W. Standart in establishing the Northern Engineering Works of Detroit, Mich., manufacturers of cranes, hoists, and foundry equipment. Since that date, he has been actively engaged with this company, as vice-president and chief engineer in charge of design and production. Mr. Reid was also an official and chief engineer of the Northern Crane Works Limited of Walkerville, Ont., manufacturers of cranes and hoists in the Dominion of Canada.

Mr. Reid was active in his interests in the Society, holding the office of secretary, treasurer, and chairman of the Detroit Section. He was also actively interested in the organization of the Associated Technical Societies of Detroit and for twenty-five years was an active member of the Detroit Engineering Society. He took a great interest in the

civic affairs of the City of Detroit and on November 22, 1922, was appointed a member of the Board of Water Commissioners of Detroit. Mr. Reid was a member of the University Club of Detroit and the Detroit Boat Club.

CHARLES E. RENSHAW

Charles E. Renshaw, president of the Dexter Watchclock Corpn. of New York, died at East Orange, N. J., on May 4, 1925. He was born in that city on May 1, 1865, and received his education at the Ashland School there. In 1882 he went to work in a clerical position with Berne, Scrymser & Co. in New York. After this concern was taken over by the Standard Oil Co. he was put in charge of the manufacture of refinery products. He was later made secretary and treasurer of the concern. In 1901 he joined the organization of Crew, Levick Co. of Philadelphia, as expert in the manufacture and application of railway lubricants. On this work he travelled through Europe, Africa, and South America in the interests of the company. In 1906 he was one of the organizers of the Newman Clock Co. and became principal owner. In 1909 he was elected president of the concern, and in 1923, by consolidating with it the Eco Clock Co. of Boston and the Hardinge Clock Co. of Chicago, he created the present Dexter Watchclock Co. Mr. Renshaw invented and patented improvements in the field of watchmen's clocks. He became an associate of the Society in 1915.

EUGENE D. ROBERTS

Eugene D. Roberts, president of the Puget Sound Iron & Steel Works of Tacoma, Wash., died in that city on August 11, 1925. Mr. Roberts was born at The Dalles, Ore., in May, 1878. He was graduated from Whitman College in Walla Walla, Wash., where his father at the time was the owner of a foundry and machine shop. Mr. Roberts received his early practical training in his father's shop, and through six month's engine-room service aboard an ocean-going vessel in 1896.

He came to Tacoma in 1898 and started in with the Puget Sound Iron & Steel Works as a draftsman. In 1903 he became chief engineer of this company and was from that time in full charge of the design of the company's products, including logging engines, logging machinery, hoisting engines, marine engines, and special equipment. He later became vice-president and eventually president of the concern, but continued to carry on his engineering work in parallel with his executive duties. Mr. Roberts became a member of the Society in 1914.

WALTER E. ROE

Walter E. Roe, general manager of production with the Dodge Mfg. Corpn., of Mishawaka, Ind., died at his home in that city on June 8, 1925. Mr. Roe was born at Nashville, Mich., on November 12, 1871, and attended school at Ann Arbor. At the early age of fourteen he entered Washington University at St. Louis, and was graduated there in 1889. For the following six years Mr. Roe was engaged in engineering work with several firms, including Allis-Chalmers Co., Swift & Co. and the Studebaker Wagon Co. He joined the Dodge Mfg. Corpn. in 1895 as chief engineer and remained in that position until 1922, when he was promoted to the managerial position which he held at the time of his death.

In addition to the great amount of development work which he accomplished upon power transmitting and conveying equipment, Mr. Roe was during the World War active in the development of airplane parts

of plywood. He carried out an extensive series of strength tests upon such parts and made them available for use in wartime planes. Mr. Roe was deeply interested in Masonic work and in 1920 was raised to the 32d degree in that order. He became a member of the Society in 1902.

WILLIAM R. RONEY

William R. Roney, a life member of the Society, died on April 6, 1925, at Annisquam, Mass., after a long illness. Mr. Roney was born in 1854 in Hamilton, Ohio. He was graduated from Chicago University where he took special courses in astronomy and civil engineering. He was one of the pioneers in the introduction of mechanical apparatus for firing coal under boilers, having invented in the early eighties the well-known stoker bearing his name. Later he became connected with Westinghouse, Church, Kerr & Co., New York City. He frequently contributed articles to the technical press. After 1910, Mr. Roney devoted his time to matters other than technical. He became a member of the Society in 1889.

H. RIALL SANKEY

Capt. H. Riall Sankey, past-president of the Institution of Mechanical Engineers (Great Britain), died suddenly on Oct. 3, 1925, at his home in Ealing, England. Mr. Sankey was born at Nenagh, Tipperary, Ireland, on Nov. 9, 1853. His early education was obtained in Switzerland and at Mr. Rippon's School at Woolwich.

While at Woolwich he was a constant visitor at the Arsenal, which aroused his interest in both engineering and military matters. Mr. Sankey entered the Royal Military Academy, where he received his commission in 1873, and from which he was sent to the School of Military Engineering at Chatham. In 1875, before his graduation there, he was one of the engineers responsible for observations upon the famous series of train tests of continuous brakes, made on the railway between Newark and Nottingham, which proved the Westinghouse system to be the best.

After graduation, Mr. Sankey was assigned to the design of barracks, soon afterward being promoted to chief draftsman of the office of the Royal Engineers at Manchester. He was then sent to Gibraltar to superintend important construction work. His next appointment was as instructor in military engineering at a school in Kingston, Canada. While there he wrote a text-book on electricity which led to his recognition by the Director of Surveys as an expert in this field.

The result was that Captain Sankey was chosen to carry out important researches at Southampton in the making of electrotypes for the printing of Ordnance maps. In his quest of a suitable high-speed engine for driving the plating dynamos, Captain Sankey came in contact with Mr. Willans, of Willans & Robinson, well-known engine builders at Thames Ditton. In 1889 Mr. Willans induced him to resign his commission and to become an engineer and director of this concern.

During the next fifteen years, which he spent with this concern, Captain Sankey invented an improved form of crankshaft, for high-speed engines, an automatic cut-off gear, steam turbines, and an improved method of affixing turbine blades.

He also made valuable applications of thermodynamics to practical engine design, and a paper by him upon The Thermal Efficiency of Steam Engines won the Telford medal and prize in 1896. Also he laid out and superintended the construction of the new Victoria works of his company at Rugby, which are still a model of their kind.

Captain Sankey resigned from the board of Willans & Robinson after the death of Mr. Willans in 1904, and for a time was a consulting

engineer at Westminster. He later joined the Marconi Wireless Telegraph Co. as consulting engineer and director, and was with them at the time of his death. Upon the outbreak of the World War, Captain Sankey returned to the army as engineering advisor to the Director of Fortifications and Works. He did splendid work in promoting the production of gages and also in evaluating commandeered industrial plants.

In 1894, Captain Sankey was elected a member of the Institution of Mechanical Engineers. He became a member of the Council in 1910 and was elected president in both 1920 and 1921. This was unusual, as there was a general rule that a new president should be chosen annually. Captain Sankey was largely responsible for the instigation of important reforms in the affairs of the Institution, among others the establishment of its Journal and its Local Branches. He then served as president of the Junior Institution of Engineers in 1922-1923. Likewise, he was an active member of the Institution of Civil Engineers, to which he was elected in 1894. Captain Sankey became a member of the A.S.M.E. in 1911, among his sponsors being his friend the late Frederick Winslow Taylor.

CHARLES E. SCHAUP

Charles E. Schaup, consulting engineer of Harrisburg, Pa., died in that city on Dec. 25, 1925. He was born on Oct. 19, 1873, at Tiffin, Ohio, received his early education in the schools of that city, and in 1897 was graduated from the Ohio State University with the degree of Engineer of Mines.

Upon his graduation Mr. Schaup entered the employ of the Lakeside & Baldwinsville Electric Railroad at Syracuse, N. Y., and was soon promoted to assistant engineer in charge of all construction work. In 1899 he became assistant engineer of the West Shore Railroad at Syracuse, and in 1900 chief engineer of the Columbus Coal & Iron Co., at Corning, Ohio. Between 1901 and 1904 he served as chief engineer of the Southern Coal & Transportation Co. and opened up a new property of 4000 tons daily coal production in Barbour County, W. Va. He then spent one year with the Central Coal & Coke Co. of Pittsburgh and Appola, Pa., and equipped their entire properties with electric power. In 1905 and 1906 Mr. Schaup was construction engineer with the Pittsburgh Filter Mfg Co. and constructed filter plants for them at Lorain, Ohio, Chilsom, Minn., and Valparaiso, Ind.

Mr. Schaup had been carrying on his private consulting work at Harrisburg, Pa., since 1908, specializing particularly upon waterworks. He had charge of numerous projects of this kind and was a recognized authority.

During the World War, he served as a Captain in the Quartermaster Corps, with headquarters at Rahway, N. J. In this capacity he had charge of the building of two Army hospitals in New York City. Mr. Schaup was an active Mason, a member of the American Water Works Association, and had belonged to the A.S.M.E. since 1916.

STANLEY D. SKILES

Stanley D. Skiles, chief draftsman in the maintenance and shop engineering departments of the Lima Locomotive Works, Lima, Ohio, died on April 25, 1925. Mr. Skiles was born on March 29, 1896, in Pittsburgh, Pa. He was educated at the Allegheny high school, and later took an Alexander Hamilton course. Mr. Skiles was first employed as a machinist with the Westinghouse Manufacturing Co., and then served as chemist at the Carnegie Steel Co. in Pittsburgh. In 1917, he became connected with the Lima Locomotive Works. Mr. Skiles became a junior member of the Society in 1922. He belonged to the Masonic Order.

ROBERT BOWEN SMITH

Robert Bowen Smith, foreman of the Development Department, Standard Oil Co. of California, died February 18, 1925, at Los Angeles. Mr. Smith was born at Chicago, Ill., October 7, 1897. He was graduated in mechanical engineering at the University of California in 1921, his college course having been interrupted by the war, during which he served for one year as a student pilot in the United States Naval Aviation Detachment at Seattle, Wash. Immediately after graduation, he began work for the Standard Oil Co., first as test operator and later as foreman. He was a junior member of the Society, having joined in 1921.

WILLIAM SNOSWELL

William Snoswell, marine and machinery inspector for the Standard Oil Co. of New York, died on March 23, 1925. Mr. Snoswell was born in London, England, in 1876, and received his education at Goldsmiths Technical College, a branch of the University of London. He served an apprenticeship at M. Bryan Donkin's while he was attending the classes at the Technical College, then went to sea as a marine engineer and obtained a first-class Board of Trade Certificate of Competency. He was foreman of a machine shop in London; also foreman of a large electric station. In 1903, he became chief engineer of several of the British and American vessels owned by the Standard Oil Co. of New York. During the course of his work for the Standard Oil Co. he had charge of the building of several large steamers. Mr. Snoswell became a member of the Society in 1922.

CHESTER M. SPALDING

Chester M. Spalding, a mechanical engineer with the General Electric Co., died at Erie, Pa., on Nov. 26, 1925. Mr. Spalding was born in Dec., 1860, in Brooklyn, N. Y. His education was obtained in the public schools, supplemented by a special course in engineering at Cooper Union.

Mr. Spalding began his mechanical work as an apprentice in the shop of Sargent & Cullingworth of New York City, where he spent two years, 1883 to 1885. After this he spent some time as a draftsman under Charles Balbridge, of Boston. In 1886 and 1887 Mr. Spalding gained further shop experience with the Superior Wax Paper Co., Newton Upper Falls, Mass.

After a short period with the Providence Steam Engine Co. he worked at the plant of the Brown & Sharpe Mfg. Co. until 1890. In that year he became a draftsman at the William A. Harris Steam Engine Co. He also did cable work in Cambridge, Mass., about that time.

Mr. Spalding joined the Hotchkiss Ordnance Co. of Providence in 1891 and soon became chief draftsman for this concern. He left them after several years to enter the employ of the General Electric Co. at Schenectady, N. Y. He was a mechanical engineer with them there and at their plant in Erie, Pa., for the remainder of his life.

Mr. Spalding became a member of the A.S.M.E. in 1894.

MAJOR HOLLON C. SPAULDING

Hollon C. Spaulding, long prominent in power-plant-engineering circles, died at New York Hospital on December 24, 1925. Major Spaulding was born in Janesville, Wis., on June 9, 1865, and received his early education at the Boston Latin School. He later entered the Massachusetts Institute of Technology, from which he was graduated

in the class of 1887 with the degree of B. S. in Civil and Mechanical Engineering.

He began his engineering career in 1888 with the Worthington Pump & Machinery Company in New York. Following this he was in charge of the motor department of the Thomson-Houston Electric Company. Between 1890 and 1896 Major Spaulding was secretary of the Thomson Vanderpoele Electric Mining Company at Lynn, Mass., and manager of the J. A. White Company at Exeter, N. H. After 1896 he was for a number of years manager of the Boston office of the Manhattan General Construction Co.

Upon the entrance of the United States into the World War, Major Spaulding entered the Army as a captain in the Ordnance Department. He served respectively as office manager of the Purchase Division of the Ordnance Department in Washington, as Officer in Charge of the Construction and Repair Division of the Army, Transport Service at Bush Terminal, New York, and finally as Port Quartermaster for the United States Army at Halifax, Nova Scotia. He received high commendation for his work at Halifax.

Before entering the Army in 1917, Major Spaulding became manager of the Contract Department of the New York Steam Company. Previous to that connection he became manager of the Hoist Department of the Yale & Towne Manufacturing Company at New York. He had also been connected with the Triumph Electric Co., as assistant sales manager.

He became a Member of the Society in 1890, and was a "Thirty-Fiver."

GEORGE HOWLAND STODDARD

George Howland Stoddard, pioneer in the modern art of refrigeration, died at his home in Boston on May 20, 1925. Mr. Stoddard was born at Plymouth, Mass., in 1855, and received his higher education at the Massachusetts Institute of Technology, from which he received the degree of Ph.D. in 1877. Almost immediately after leaving college Mr. Stoddard became interested in the possibilities of mechanical refrigeration, which was then in the laboratory stage of development. His first venture in this field was in the organization of the Mechanical Refrigerating Co. of Boston, of which he was engineer and manager. This company did not make a success of the new art but did pave the way for later success. In 1889 Mr. Stoddard organized the Faneuil Hall Cooling Co. of Boston. Within a year this became the Quincy Market Cold Storage and Warehouse Co., now one of the largest refrigerating projects in the world. Mr. Stoddard designed most of the original equipment for this plant, was its manager and engineer, and at the time of his death was treasurer of the company.

On account of his valuable research and his practical constructive work Mr. Stoddard was selected to represent the American Association of Mechanical Refrigerating Engineers at the international convention in Paris about ten years ago. He became a member of the Society in 1897.

HOMER H. STUART

Homer H. Stuart, industrial engineer with the De Laval Separator Co. of Poughkeepsie, N. Y., died at the Vassar Hospital in that city on August 29, 1925, after an illness of three months. Mr. Stuart was born at Athens, N. Y., on July 4, 1890, but spent his boyhood in Philadelphia, where he attended the public schools. He was graduated from

the Bordentown Military Academy and then entered Lehigh University, where he studied mechanical engineering for three and a half years.

Upon leaving the University Mr. Stuart went to the plant of E. & T. Fairbanks & Co., St. Johnsbury, Vt. At first he spent a year and a half in their shop-training course, during which time he worked in all departments of the shop, in the drafting room, and at the New York office, where he was engaged in plan work. Upon the completion of this course in 1917 he was appointed assistant to the factory superintendent at St. Johnsbury. In 1918 he was in addition made head of the Claims Department.

Mr. Stuart left the Fairbanks Co. in 1919 to join the engineering staff of the De Laval Separator Co. He was a Mason, a member of the Kappa Alpha Society, and had belonged to the Society since 1917.

JOHN M. SWEENEY

John M. Sweeney, recognized as an authority on boat construction for inland waterways and a member of the Society for the past forty years, died on April 13, 1925, at his home in Hollywood, Cal. Mr. Sweeney was born on July 10, 1851, in Wheeling, W. Va. He was graduated from Washington and Jefferson College in 1869 and shortly thereafter joined his father in the firm of A. J. Sweeney & Sons Co., engaged in foundry, machine-shop, and shipbuilding work. While affiliated with this company, Mr. Sweeney assisted in designing and building many large inland-waterway steamboats. From 1900 to 1920 he had offices in Chicago where he was actively engaged in consulting work, making a specialty of marine designs and contracts. He was also connected with the Wheeling Electric Co., the Wheeling Bridge Co., and the Wheeling Railway Co. During the World War he built ships for the Government at Pensacola, Fla., where he was president of the Pensacola Ship Building Co. For the past five years Mr. Sweeney has devoted his time to writing, contributing many articles relating to marine craft to the *Waterways Journal* and similar publications. He was a member of the Western Society of Engineers and served on its membership committee.

J. HAWLEY TAUSSIG

J. Hawley Taussig, gas engineer of the Sales Department of the United Gas Improvement Contracting Co., Philadelphia, Pa., died on March 2, 1925, following an operation. Mr. Taussig was born at Washington, D. C., in January, 1876. He was a son of the late Rear Admiral Taussig, U. S. N. Following his graduation from Cornell University, in 1897, he became foreman of various plants of the Philadelphia Gas Works and did special engineering work on boilers, engines, and water systems for the engineer of works. In May, 1901, he was made superintendent of the Market Street Gas Works, Newark, N. J., and in 1904 went to the United Gas Improvement Co. at Philadelphia, as assistant engineer. Later he became engineer of design, and designed and developed a new system of vertical gas retorts. He also designed waste-heat boilers, vertical retorts for gas making, gas take-off connections and take-off cleaners. A few years ago the Edward Longstreth Medal was awarded to him by the Franklin Institute for his invention.

Mr. Taussig became a member of the Society in 1916, and belonged also to the American Gas Association, the National Gas Association of America, and the Franklin Institute.

JOHAN O. E. TROTZ

Johan O. E. Trotz, well known as a metallurgical engineer both in the United States and in Europe, died at his home in Savnevik, Ronneby, Sweden, on July 18, 1925. Mr. Trotz, who was born in Stockholm in 1860, was graduated from the Royal Technical High School and the Royal School of Mines in that city. For a year and a half after his graduation from the latter institution he remained there on special research for the Association of Swedish Iron Masters. Upon completion of this work he became superintendent of blast furnaces and Bessemer steel works at the Iggesund Iron & Steel Works. He was also for a year superintendent of open-hearth-steel furnaces, puddling furnaces, and iron foundries for the Surahammar Iron & Steel Works.

Having in the meantime been awarded scholarships by the Association of Swedish Iron Masters and by the Swedish government, Mr. Trotz next gave up a considerable period to the study of the iron and steel industry in Germany and in the United States. He remained in the United States for many years, fifteen of which were spent with the Washburn & Moen Mfg. Co. and its successor, the American Steel & Wire Co. He started in as chemist and inspector of materials and became general superintendent of their South Works and Central Works in Worcester, Mass. Mr. Trotz was later appointed metallurgical engineer of the United States Steel Corpn. with headquarters at the president's office in New York City. He was chairman of their open-hearth committee and as such inspected the leading steel plants in this country and those in England, France, Italy, Austria, Germany, and Sweden.

In 1909 Mr. Trotz returned to Sweden where he became a consulting metallurgical engineer and inventor. At the time of his death he was managing director of Kockums Jernverks, Aktb. of Kallinge. Mr. Trotz became a member of the Society in 1906.

LYMAN A. UPSON

Lyman A. Upson, manufacturer and financier, died at Thompsonville, Conn., on January 21, 1924. Mr. Upon was born at Westfield, Mass., July 23, 1841, was graduated from the Massachusetts State Normal School in 1860, and upon the outbreak of the Civil War enlisted in the 62d New York Infantry. He served throughout the War in the Army of the Potomac, going through the battles of Fredericksburg, Gettysburg, and the Wilderness.

After being mustered out of the service, Mr. Upson went to work for the Hartford Carpet Co., which is now the Bigelow-Hartford Carpet Co. He remained with this concern for thirty-three years. In 1901 he resigned as assistant to the president in order to organize the Upson-Martin Co., carpet manufacturers at Thompsonville. He continued as president and treasurer of this concern up to the time of his death, at the same time serving as president of the Thompsonville Trust Co.

Mr. Upson was the designer of several textile mills and warehouses, and specialized in the installation of motor drives. He supervised the installation of the Enfield Electric Light & Power Co., of which he was president, and the Enfield & Longmeadow Electric Railway, of which he was secretary and treasurer. He was the inventor of a machine for cutting pattern cards for Jacquard looms, a device for lacing these looms, and also of a machine for the continuous scouring and drying of yarn.

Mr. Upson joined the Society in 1884 and was one of the last of the members of the Grand Army of the Republic in the Society. He was a charter member of the Engineers' Club of New York.

J. HENRY VAILE

J. Henry Vaile, for many years known as an expert in the design of pumping and oil machinery, died at his home in Dayton, Ohio, on May 8, 1925. Mr. Vaile was born on March 31, 1844, at Piqua, Ohio. He attended high school at Lowell, Mass., and then entered upon a three years' apprenticeship as a machinist in the old Lowell Machine Shop. During the Civil War he enlisted in the U. S. Navy and attained the rank of Second Assistant Engineer.

After the war, Mr. Vaile became an erecting engineer with the Holley Mfg. Co. at Columbus, Ohio. He left them to become engineer in charge of the pumping works of the Indianapolis Water Co. During this period he had charge of extensive street and canal work in Indianapolis. In 1874 Mr. Vaile became vice-president and superintendent of the Smith & Vaile Co., and from that time forward he specialized upon waterworks and cotton oil machinery in the main. He was subsequently president of the National Waterworks Construction Co., and had an active part in the building of the waterworks at Raleigh, N. C. At the time of his retirement several years ago, Mr. Vaile was president of Vaile & Kimes of Dayton, Ohio. He became a member of the Society in 1888.

WILLIAM C. WAY

William C. Way, supervisor of the Surveys Laboratory, Department of the Interior, Ottawa, Can., died at Lindsay, Ont., on January 16, 1924. Mr. Way was born at Belleville, Ont., on October 10, 1881, and received his early education in the school and academy there. He showed a great fondness for science and obtained a scholarship at Queen's University, where he specialized upon engineering. He received a degree of B.Sc. in 1905, an M.Sc. degree in 1906, and also a diploma in civil engineering. Mr. Way began his practical work in 1905 as town engineer of Lindsay, Ont. Later he was for a time with the General Electric Co. in Lynn, Mass., and with the Allis-Chalmers Co. in Montreal, both in the drafting and testing departments. He also spent some time as sales engineer for Canadian Ingersoll-Rand. Following this he was for two years lecturer in mechanical engineering at Queen's University.

Mr. Way became supervisor of the Canadian Surveys Laboratory in 1912. Here he had charge of the government standards of length, weight, time, temperature, etc., and calibrated all sorts of optical instruments and other instruments of precision. Mr. Way never was in robust health, yet he gave unlimited energy to his duties and went far beyond them in his researches. He became a member of the Society in 1921 and was also a member of the Engineering Institute of Canada.

CHARLES T. WESTLAKE

Charles T. Westlake, chief mechanical engineer of the Commonwealth Steel Co., died at his home in St. Louis, Mo., on July 22, 1925. He was born at Syracuse, Mo., in June, 1864, and was educated in the common schools. At the age of 17 he began an apprenticeship at the Western Iron Boat Building Co., St. Louis, serving as carpenter, pattern maker, machinist, and draftsman. His first engineering experience was upon the design of snag boats for the Mississippi River Commission. After this apprenticeship he worked for seven years at the Westlake Button & Novelty Works upon contractors' equipment and special machinery.

In 1895 Mr. Westlake designed and installed a complete sewer system with pumping station for Granite City, Ill. He also served for a

time as chief engineer at the plant of the Commonwealth Steel Co. there. He later redesigned and doubled the capacity of the plant of the American Steel Foundry Co., where he built and operated a car shop for the manufacture of steel freight cars of his own design. Mr. Westlake was the holder of about forty patents, mostly covering improvements upon railway equipment, but also including a unit sleeve valve motor. Among his railway inventions may be mentioned the novel freight car truck, manufactured by the Swing Rolling Truck Co., of St. Louis, of which concern he was president. Mr. Westlake became a member of the Society in 1910.

SETH WHEELER, SR.

Seth Wheeler, Sr., president of the A. P. W. Paper Co., and recognized internationally as an authority on paper, died on May 22, 1925, at his home in Albany, N. Y. Mr. Wheeler was born on May 18, 1838, in Chatham, N. Y. He was educated in the district schools and in the Albany Academy. In 1849 the Wheeler, Melick & Co., of which his father was head, moved to Albany where until 1872 Seth Wheeler acted as mechanical superintendent. He resigned in that year to devote his time to the development of machinery which later gave him fame as an inventor and led to the organizing of the A. P. W. Paper Co. His inventions for perforating and winding paper on rolls are used throughout the world and the company is now parent to many other companies established in this country and abroad.

Mr. Wheeler was for many years president of the Albany County Savings Bank and since his retirement from that office has served continuously as a director of that institution. He was also a director of the New York State National Bank. He became a member of the Society in 1894. He belonged to the Fort Orange Club and the Masters Lodge of Masons.

JAMES T. WIKLE

James T. Wikle, head of the Mechanical Department and purchasing agent for the Fulton Bag & Cotton Mills, of Atlanta, Ga., died on February 18, 1925. Mr. Wikle was born at Cedartown, Ga., in 1876, and was graduated from the Georgia School of Technology in 1896. Almost immediately after graduation, he entered the employ of the Fulton Bag & Cotton Mills. He started as draftsman, then was promoted to be assistant superintendent, and for the past twenty-five years was head of the Mechanical Department, looking after all construction work, repairs of machinery and plant, and of power. He also supervised the purchase of all equipment and supplies, and the designing and building of a number of special machines. Mr. Wikle became a member of the Society in 1917, and during the winter of 1923-24 was chairman of the Atlanta Local Section.

WILLIAM H. WILEY

William H. Wiley, Treasurer of The American Society of Mechanical Engineers since 1884, died in his eighty-third year at his home in East Orange, N. J., on Saturday, May 2, 1925, after an illness which started about the middle of January. He had been identified with the publishing world since 1876, in which year he joined his father, John Wiley, and his older brother, Charles, doing business under the firm name of John Wiley & Sons.

Major Wiley was born in New York City on July 10, 1842, where he spent the first dozen years or so of his life. In the early fifties the

family moved to East Orange, N. J. Major Wiley was attending the College of the City of New York when the Civil War broke out. He immediately joined the Independent Corps of the New York Volunteer Light Infantry as first lieutenant, an organization later known as the Seventh Regiment. Many years afterward he was elected president of the Seventh Regiment War Veterans.

In the course of the war Major Wiley rose to the command of two companies of artillery, seeing most of his service in South Carolina, where he served with distinction in a number of severe engagements. He was given special mention by General Gillmore "for efficient and able services during the siege and bombardment of Fort Sumter and Charleston, S. C." In 1864 he was retired as brevet major "for gallant and meritorious services."

After the war Major Wiley resumed his studies and in 1866 received the degree of civil engineer from Rensselaer Polytechnic Institute, following this course with a year of special work at the Columbia School of Mines. For the next nine years he practiced engineering in the East and Midwest, dropping this practice to enter the publishing business in 1876. The Wiley publications at that time included books of many sorts, but Major Wiley soon saw a great opportunity to develop the business into one of technical and scientific books. From his service in the war and as an engineer he noticed the great dearth of good American scientific books. His friendship with officers of both the Army and Navy made it natural that he should be selected both by West Point and Annapolis as the publisher of their books.

Major Wiley's wide and varied experience in the engineering field brought him into direct contact with the most prominent engineers of the country and gave him a first-hand knowledge of their needs. This knowledge enabled him to make many suggestions which led to the writing of needed engineering books by well-qualified authors. Thus the business became definitely one of a scientific nature.

Under his able direction, which continued until the last year of his life, the Wiley lists grew to include books in all the branches of engineering, agriculture, chemistry, geology, mathematics, and the biological sciences. In the last few years of his life, realizing the close link between engineering and business, he encouraged his associates to develop a group of books in the latter field.

Despite his great activity in the publishing field, Major Wiley found time for many other activities. For three years he was a member of the township committee of East Orange, serving for one year as its chairman. He served three terms in Congress as a representative from New Jersey. He was a commissioner from New Jersey to the St. Louis Exposition in 1904 and was president of the International Jury of the Brussels Exposition in 1907, being decorated with the Order of Leopold by the King of the Belgians. He was a member of the executive committee and during the World War was chairman of the National Preparedness Committee of The American Society of Mechanical Engineers. He was a member of the American Society of Civil Engineers, the American Society of Mining and Metallurgical Engineers, American Institute of Electrical Engineers, and The American Society of Mechanical Engineers. He belonged also to the American Association for the Advancement of Science. Major Wiley was a member of many clubs, including the University and Engineers' Clubs of New York, the Cosmos Club of Washington, and many others.

Major Wiley was one of the eighty men present at the meeting on March 15, 1880, at which The American Society of Mechanical Engineers was founded. Four years later he became the treasurer of the Society, the third to hold that office, which was his to the time of his death, a record of forty-one years of unparalleled service to the organization.

In a word of personal appreciation, Dr. D. S. Jacobus, past-president of The American Society of Mechanical Engineers, writes as follows:

"He was a man of close and strong friendships. A home lover. To him it mattered little what others thought. There was always that chivalry and consideration which bespeaks a true gentleman. Above all, he had the quality that convinced one of absolute truth and frankness which, with his kindly nature, endeared him to all. In the early days of the Society he was ever ready to lend a helping hand, and he smoothed out and remedied many an unpleasant situation. His earnest efforts and foresight contributed much to the upbuilding of our Society and he stands out as a typical example of the pioneers who have given freely of their best efforts to make it what it is today."

Mention should be made of the work that Major Wiley did in the early days of reporting the meetings of The American Society of Mechanical Engineers in London *Engineering*. These reports were models for accounts of the sort and did much to bring the work of the Society before the world at large. As an example of these reports may be cited the descriptive accounts which he wrote of the A.S.M.E. Meeting held in Washington, D. C., May 9 to 12, 1899, and which were published in six large installments in issues of *Engineering* running from July 7 to August 11 of that year.

KENNETH F. WOOD

Kenneth F. Wood, treasurer of the Sayles Finishing Plants, Inc., died at his home in West Barrington, R. I., on September 22, 1925. He was apparently recovering from an attack of pneumonia when he passed away very suddenly. Mr. Wood was born at Central Falls, R. I., on May 24, 1873. He received his early education at Cole's Private School in Pawtucket, then entered the Massachusetts Institute of Technology, from which he was graduated in 1894 with the degree of B.S. Immediately upon graduation he went to the Sayles Finishing Plants at Saylesville, R. I., as a draftsman. Mr. Wood's progress here was rapid, and he became successively master mechanic, superintendent, general manager, and treasurer and a director. It is worthy of mention that Mr. Wood was closely associated with both Frederick W. Taylor and H. L. Gantt in their earlier work in industrial efficiency. Besides his major activities at the Sayles Plants, Mr. Wood was a director of the following corporations: Rhode Island Hospital Trust Co. of Providence, Chase National Bank of New York, Hamlet Textile Co. of Woonsocket (V. P.), The Wauregan Co. of Wauregan, Conn. (Pres.), the Slater Yarn Co., Crown Mfg. Co. and Union Wadding Co. of Pawtucket, the Moshassuck Valley Railroad of Saylesville, Ponemah Mills of Taftville, Conn., and the American Bleached Goods Co. of New York. Mr. Wood also gave up much time to philanthropic causes. He was a member of Delta Upsilon Fraternity and of the A.S.M.E., which he joined in 1899.

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